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USAAVLABS TECHNICAL REPORT 65-52

A MAIN POWER SYSTEM FOR SHAFT-DRIVEN HEAVY LIFT HELICOPTERS

By

Robert B. Bossler, Jr.

October 1965

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-212(T)
KAMAN AIRCRAFT CORPORATION

DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA 23604

This report represents a part of the program being conducted by the U. S. Army Aviation Materiel Laboratories to investigate mechanical transmission system concepts for a shaft-driven heavy-lift helicopter of the 75,000-pound to 95,000-pound gross weight class. The purpose of this investigation was to determine the high risk or problem areas that could be expected in the development of a drive train for a mechanically driven heavy-lift helicopter.

This report presents a comparative analysis of two power train configurations utilizing a gas-coupled power turbine driving a coaxially mounted main reduction transmission.

This command concurs with the contractor's conclusions reported herein.

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A MAIN POWER SYSTEM FOR SHAFT-DRIVEN HEAVY LIFT HELICOPTERS

Kaman Report No. R-555

By

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Prepared By

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For

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Fort Eustis, Virginia

SUMMARY

A study was conducted to optimize a geared power system for a heavy lift helicopter (HLH), and to determine specific areas requiring maximum development effort, specific weights, and complexity. The power transmission concept explored here uses multiple gas generators, conventionally installed, and gas-coupled to a peripherally-driven remote turbine of the lift and cruise fan type. The remote turbine is co-axially mounted to a speed-reducing gearbox which is also co-axial with the rotor. This concept is known as the turbine integrated geared rotor (TIGR). Two speed-reducing gearbox configurations are considered. The first uses an integrated rotor hub/ring gear in which the large diameter rotor hub required for HLH has an internal ring gear as its inner surface with the main hub mounting bearings above and below the gear. The second speed-reducing gearbox has the rotor hub segregated from the gearbox. The configurations are extremely compact and necessarily affect many adjacent systems performing essential functions for HLH. Accordingly, a three-part program of design, analysis, and comparative evaluation was undertaken which included the effect on adjacent systems and the resulting HLH aircraft.

The design for this power transmission concept encompasses the transmission; rotor hub, rotor blade retention, articulation, and control; affected portions of the fixed and rotating controls; affected portions of the hydraulic and electrical systems; affected portions of aircraft structure; accessory and tail rotor drive provisions; tail rotor drive by hypercritical shafting; and the power plant.

The analysis phase includes manufacturing technology, weight, efficiency, stress, deflection, dynamic behavior, reliability, maintainability, cost, and development time.

The comparative evaluation of the mechanical drive systems includes parametric analysis to define total aircraft, comparison of performance of identical missions by total aircraft, comparison of total aircraft, and evaluation.

The results show the TIGR concept to be superior to conventional multiple engine/transmission practice for HLH with respect to the essential factor of reliability. The segregated rotor hub with fixed ring gear was found superior to

a rotating ring gear integral with the rotor hub. The TIGR concept is compatible with present conceptions of HLH aircraft.

No maximum effort development areas were found. The fact remains, however, that interactions between the closely juxtaposed rotor, gearing, and turbine may introduce problem areas that cannot be predicted by paper studies alone. It is recommended, therefore, that this novel concept be investigated by power system testing concurrent with additional studies for better HLH definition.

FOREWORD

This report covers a design study conducted from 1 July to 30 December 1964, to determine specific areas requiring maximum development effort, specific weights and complexity of two power transmission systems for shaft-driven, heavy lift helicopters (Contract DA 44-177-AMC-212(T)). Kaman Aircraft Corporation was the prime contractor, the principal investigator was Mr. Robert B. Bossler, Jr., and the contract administrator was Mr. Walter C. Kenyon, Jr. Mr. A. B. Jones was engaged as a bearing consultant. Mr. Jones is a nationally known specialist in the analysis of rolling element bearings. The method of analysis which he derived for design of the large integrated hub bearing is presented in the appendix to this report.

Design and analysis of gas generators, ducting and turbine was performed by the General Electric Company's Small Aircraft Engine Department at no cost to the contract. Because of the closely-integrated nature of this propulsion system concept, General Electric personnel worked as tightly coordinated members of the engineering team throughout the course of the program.

The gearing involved in heavy lift helicopter transmission is much larger than is familiar to the helicopter industry. In order to assure an experienced, realistic appraisal of manufacturing feasibility, therefore, a subcontract was concluded whereby the transmission designs were reviewed and analyzed by the King of Prussia Research and Development Corporation, a subsidiary of the Philadelphia Gear Corporation. The Philadelphia Gear Corporation has produced gears of larger size and greater accuracy than required for HLH. They supplied specialized information on large gearing technology and performed analytical work as directed.

Analysis and weight estimation of the hypercritical tail rotor shafting was performed by the Utica Division of the Bendix Corporation, based on their experimental extension of research conducted at Battelle under Army sponsorship.

The Government Representatives at U.S. Army Aviation Materiel Laboratories (formerly U.S. Army Transportation Research Command), Fort Eustis, Virginia, were Mr. R. P. McKinnon, contracting officer and Mr. W. A. Hudgins, project engineer.

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LIST OF SYMBOLS

A_b	total rotor blade area
APU	auxiliary power unit
BTU	British thermal unit
DGW	design gross weight
f	flat plate drag area
FPM	feet per minute
g	gravitational acceleration
GW	gross weight
HLH	heavy lift helicopter
HP	horsepower
MIL	military power rating
MQT	military qualification test
NRP	normal rated power
OGE	out-of-ground effect
PFRT	preliminary flight rating test
POL	petroleum, oil, lubricants
R	radius
R/C	rate of climb
SFC	specific fuel consumption
SHP	shaft horsepower
SL STD	sea-level standard
TBO	time between overhaul

LIST OF SYMBOLS (Continued)

TIGR	turbine integrated geared rotor
USAAVLABS	U. S. Army Aviation Materiel Laboratories
V_t	rotor blade tip speed
W	weight
ρ/ρ_o	altitude density/sea-level standard density

INTRODUCTION

Earlier design studies of HLH showed tip-driven rotors provided higher payload/gross weight ratios than mechanically-driven rotors for the Army missions then specified. Fuel efficiency was low, but this was overbalanced by the high weight and complexity of existing helicopter mechanical drive systems. The high fuel efficiency of mechanically-driven rotors for HLH is extremely attractive, however, especially in view of the fuel logistics problem in remote areas such as Vietnam. Weidhuner states, "It has been variously reported that 40 percent to 60 percent of the total support of an overseas Army is petroleum products" (reference 12, page 7). Two recent developments in power transmission technology tip the scales in favor of mechanically-driven systems for HLH.

The first development is the demonstrated practicality of using multiple gas generators, gas-coupled to a remote turbine as in lift and cruise fans. It is worthy of note that large electric power generators are driven by turbines using remote multiple gas generators at power levels up to ten times larger than required for the HLH considered here. The geometry of the multiple gas generator/remote turbine arrangement selected for the HLH eliminates from transmission design the functions of engine combining, change of direction, part of the required speed reduction, considerable shafting, and all of the multiple individual-engine, overrunning-clutch provisions required for engine-out operation and for autorotation.

The second recent development in power transmission technology that reduces weight and complexity is the development of techniques to manufacture hardened and ground gears of large size, low diametral pitch, and required accuracy. The capability extends to drive trains more than three times as large as required for the HLH considered here. The transmission designs reflect these advances in metallurgy and manufacturing technology with significant weight/size reduction.

Design study of HLH was undertaken based on these two developments, resulting in the TIGR concept. A contract for this study program was awarded following a competition for design studies in this area. This report presents the results of this study.

PROGRAM DISCUSSION

A. HLH DESIGN CHARACTERISTICS AND MISSIONS

(As quoted from the contract work statement)

1. Design Characteristics

(a) Gross weight - 75,000 to 85,000 pounds
(b) Turbine powered
(c) Safe autorotation at design gross weight
(d) Design load factor of 2.5 at design gross weight
(e) Crew: minimum of 1 pilot, 1 copilot, and 1 crew chief
(f) All components to be designed for 1200 hours between major overhaul and 3600 hours service life

2. Missions

(a) Transport mission

Payload:	12 tons (outbound only)
Radius:	100 nautical miles
Vcruise:	12-ton payload, 110 knots
Vcruise:	no payload, 130 knots
Hovering time:	3 minutes at takeoff, 2 minutes at midpoint with payload
Reserve fuel:	10 percent of initial fuel
Hover capability:	6000 feet, 95°F (out-of-ground effect), takeoff gross weight
Mission altitude:	sea-level standard atmosphere
Fuel allowance for start, warm-up, and takeoff:	MIL-C-5011A

(b) Heavy lift mission

Payload:	20 tons (outbound only)
Radius:	20 nautical miles
Vcruise:	20-ton payload, 95 knots
Vcruise:	no payload, 130 knots
Hovering time:	5 minutes at takeoff, 10 minutes at destination with payload
Reserve fuel:	10 percent of initial fuel
Hover capability:	sea level, 59°F (out-of-ground effect)

Mission altitude:	sea-level standard atmosphere
Fuel allowance for start, warm-up, and takeoff:	MIL-C-5011A

(c) Ferry mission

Ferry range (no payload, STOL takeoff):	1500 nautical miles
Reserve fuel:	10 percent of initial fuel
Fuel allowance for start, warm-up, and takeoff:	MIL-C-5011A
Minimum design load factor:	2.0
Mission altitude:	sea-level standard atmosphere
Speed:	best for range

B. POWER SYSTEM SPECIFICATIONS

1. Defined

The following specifications were established by the request for quotation and the prebid briefing and had a direct effect on design:

(a) During the design study, reliability shall be considered second only to performance of the aircraft mission. Further consideration shall be given to maintainability, with estimates as to the complexity of the HLH transmission system.

(b) Design studies shall consider the integration of the rotor control system with the power transmission system. Controls shall be within the shaft, if feasible.

(c) Transmission and rotor system must be capable of transferring full power capabilities of power plant at standard sea level conditions.

(d) A rotor brake is required for engine check-out with rotor locked.

(e) Contemplated fuel is JP4.

(f) For estimating latest technology available, time frame for HLH development will probably be 1968-1972. Part of the studies will determine areas requiring development prior to full development program.

(g) Desirably, aircraft accessories will be operative by APU and main drive.

(h) All cargo will be interior except on heavy lift outbound.

(i) Complete the transport mission, with exception of original hover, with one engine out.

(j) The three missions are of equal importance.

(k) For performance calculations, use power required versus velocity curve (see Figure 2 and Figure 3).

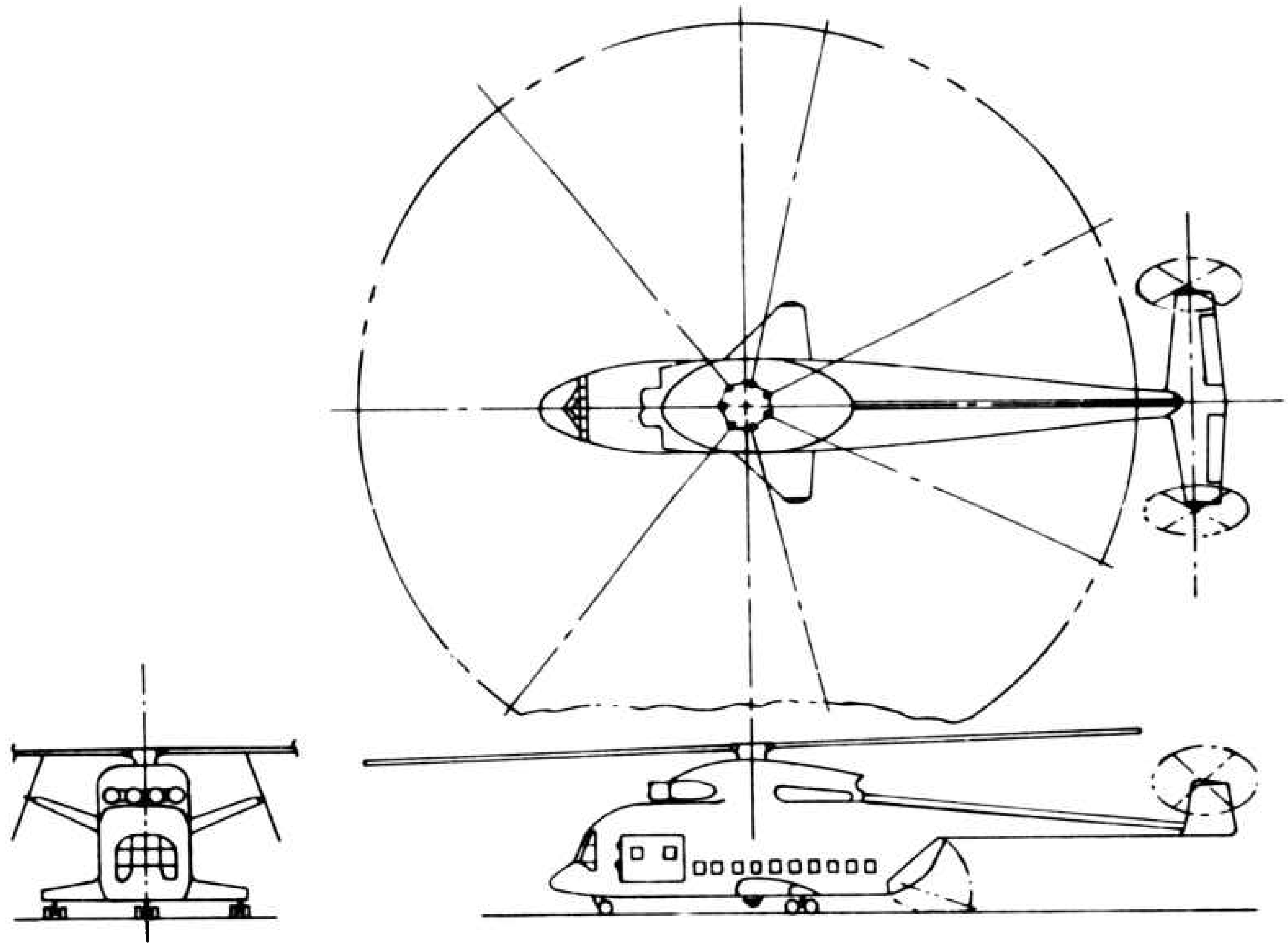


Figure 1. Heavy-Lift Helicopter - Internal Cargo

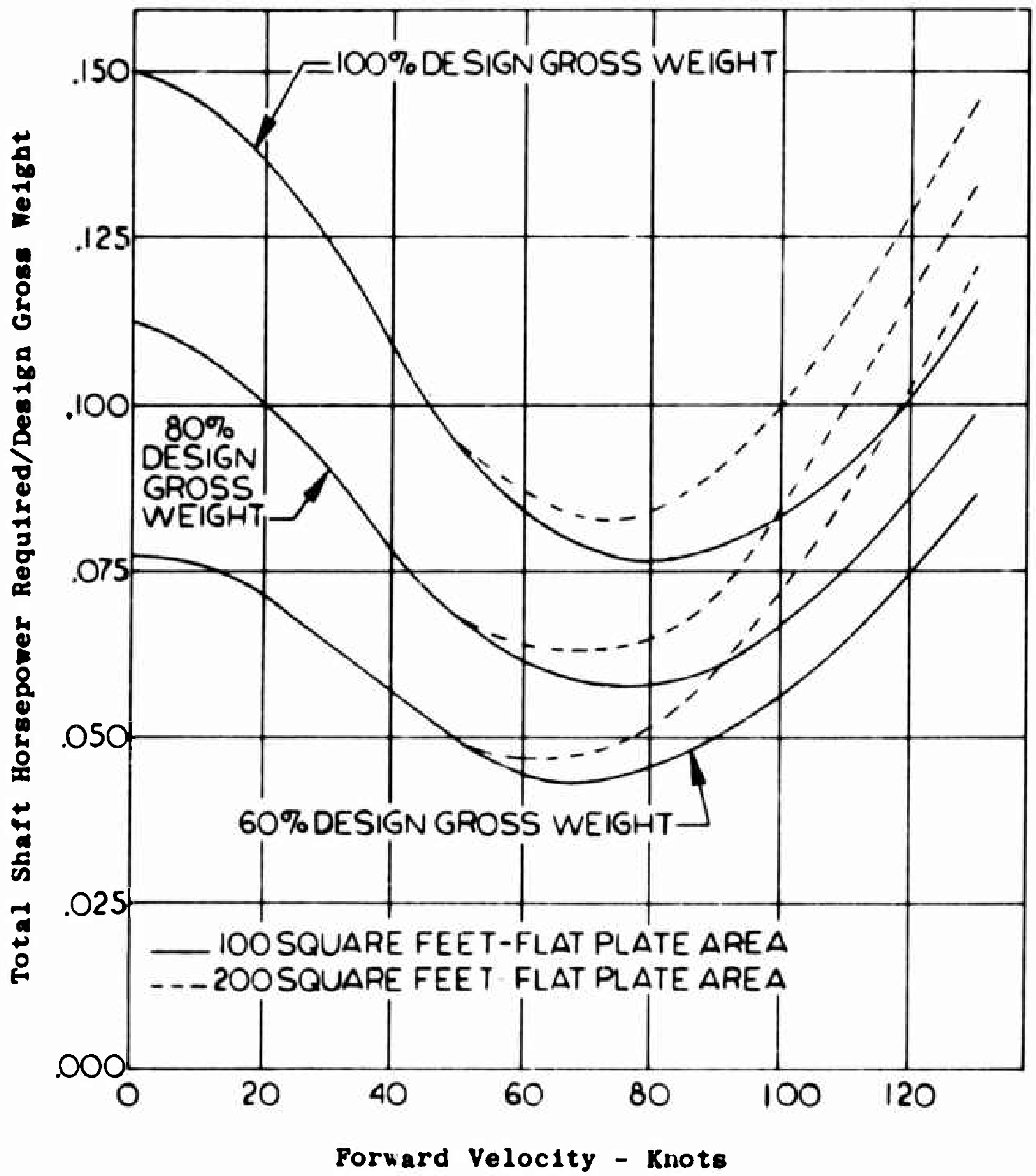


Figure 2. USAAML Power Required Curve
Shaft Horsepower/Design Gross Weight
Versus Forward Velocity

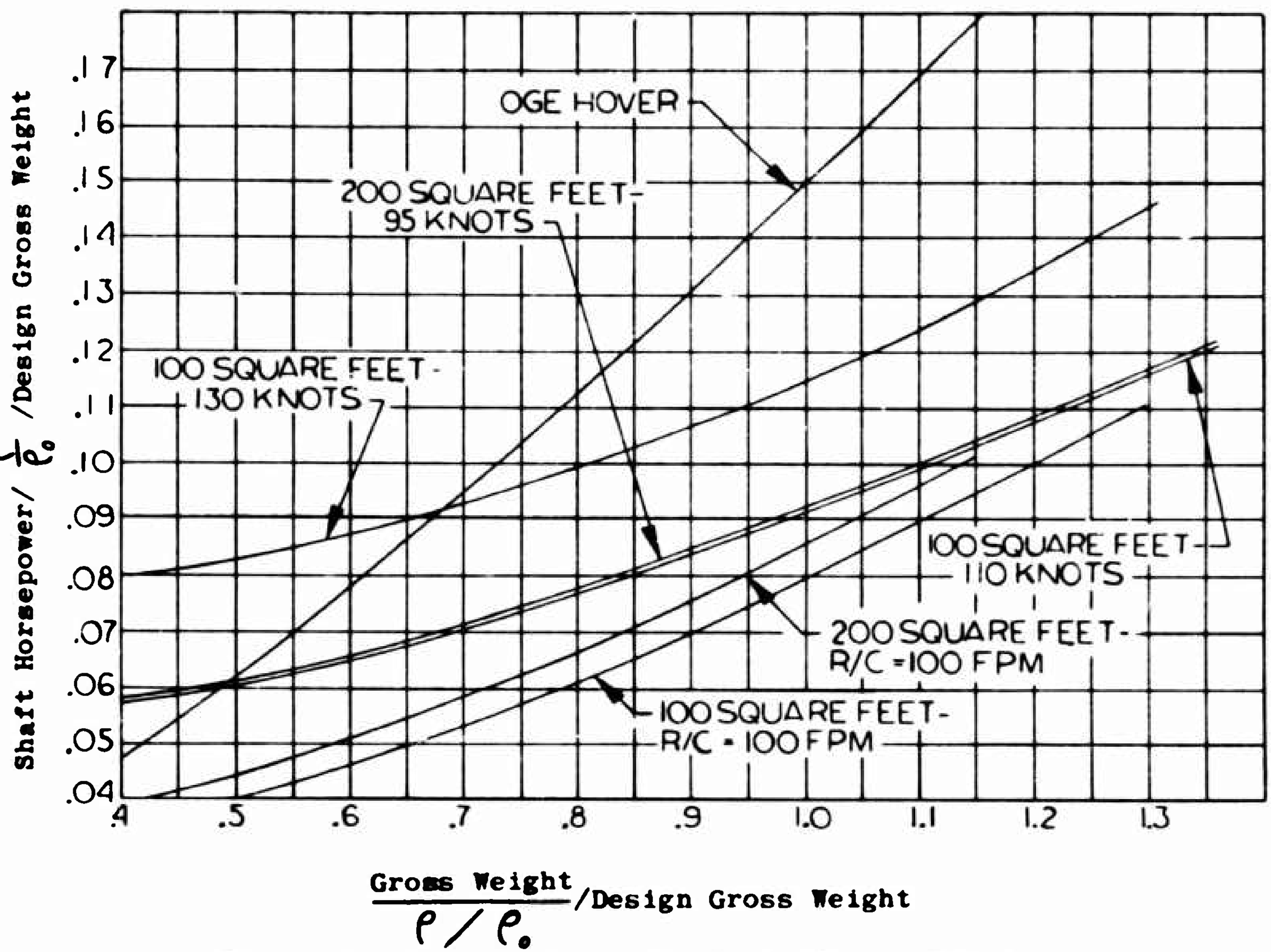


Figure 3. Mission Power Required (Figure 2 Data)

2. Derived

The power system design will be based on information available from preliminary analysis. Following design, the analysis will be re-entered to correct for calculated power system weight. Parametric analysis will be used to determine revised HLH component and system weights and HLH power required. Power system design weight will be scaled up or down consistent with the new power required, and then the entire analysis recycled until convergence occurs.

The transmission will be designed initially for 12000 horsepower, 124-r.p.m. output speed, based on preliminary estimates of HLH gross weight and system parameters.

The power system speed and power specifications were derived from consideration of the USAAML* definition of HLH design characteristics, missions, and power system specifications. The initial step was to identify rotor characteristics consistent with the specified power and to select appropriate engines. Techniques were derived to relate design gross weight, operating gross weight, weight empty, fuel required, and Army hot day power. USAAML hover data may be matched approximately by a rotor with a 90-foot diameter, 650-f.p.s. tip speed, and 0.135 solidity. The high solidity is required to meet the airspeed requirements at this high disc loading. Such a rotor has a low efficiency as a lifting device and requires four T64-S5A gas generators to drive the remote turbine rather than the three T64/S5A gas generators originally contemplated to be used with a rotor of 100-foot diameter, 650-f.p.s. tip speed, and solidity of 0.12. The 650-f.p.s. tip speed, required to meet the speed/drag parameters specified, fixes transmission output speed at 124 rotor r.p.m. for the more efficient lifting rotor originally selected. The change from three to four gas generators increased ship empty weight and gross weight. Using the new gross weight and the USAAML power required data, the multiple gas generator/remote turbine engine was then fixed at 12000 horsepower sea-level standard, flat rated. The power delivered to the gearbox by four T64/S5A gas generators at 6000 feet, 95°F is 11350 horsepower MIL. This engine meets USAAML power requirements.

The HLH configuration selected from the above criteria can overperform all missions, or can perform all missions with at least one inoperative gas generator. Also, it appears that the resulting HLH could perform all missions with four T64/S4A gas generators, although the USAAML power-required

*Changed to USAAVLABS in June 1965.

specification might not be met. With four S4A gas generators, the multiple gas generator/remote turbine engine has the same sea-level standard 12000 horsepower, and is also flat rated. MIL power at 6000 feet, 95°F is 9140 horsepower as delivered to the gearbox, based on the 2450-horsepower MIL rating for the S4A under these ambient conditions. For the 2650-horsepower takeoff rating, proposed for another program, takeoff power is 9890 horsepower at 6000 feet, 95°F. With efficient rotor design, these horsepower ratings will provide more than a desirable 10 percent power margin in all missions. The following considerations guided the decision to use four gas generators rather than three; and to accept the consequent weight/cost penalty.

(a) This study can be compared directly to the two other USAAML-sponsored transmission studies, because the same power-required data will be used in all three studies, thus accomplishing this USAAML purpose.

(b) The HLH missions cannot be precisely defined until actual experience is accumulated. Therefore, the ability to overperform all missions appears to be desirable.

(c) The lower power T64/S4A gas generator will be available before the higher power T64/S5A. The development time for the HLH can be shortened by a design approach wherein four S4A gas generators can be used initially, accomplish the specified missions immediately, and allow the growth potential inherent in the subsequent change to the growth higher power S5A gas generators.

(d) It was noted that USAAML plans design studies entitled parametric analysis and preliminary design of a shaft-driven rotor system for heavy lift helicopters (Neg 94) and cost effectiveness study of a heavy lift aerial vehicle (Neg 93). Changes in power required resulting from these studies may be in either direction, but are not expected to require more than four gas generators. Selection of four for design purposes will therefore give an accurate or conservative transmission weight, will be adequate to support the planned studies, and is valid for the purpose of comparing two speed-reducing transmission concepts.

Transmission will be designed for 3600-r.p.m. input speed. Turbine data furnished by an engine manufacturer were analyzed, resulting in a selection of 3600 turbine r.p.m. on the basis of trade-off between gearbox reduction ratio required, gearbox weight, turbine weight, and system losses converted to effective weight. The 3600 r.p.m. is transmission input r.p.m.

Seven rotor blades will be used. The superposition of rotor and transmission loads on the integrated rotor hub/ring gear is the key factor requiring resolution. To reduce local blade loads, and hence local deflections of the rotor hub/ring gear, a relatively large number of blades is desirable. Rotor blade structural considerations with the selected tip speed, solidity, disc loading and airfoil show seven blades to be satisfactory with respect to performance and to the ratios of blade length to chord and blade length to thickness. The rotor study may find five or six blades to be preferable. This can be accommodated at a later date. The determination of an optimum number of blades is beyond the scope of the present study.

The accessory gearbox will be separate from the main transmission and be located aft of the main transmission. The accessory gearbox will be driven by the tail rotor drive shaft, as well as by an auxiliary power unit.

An auxiliary power unit (APU) will power the accessory gearbox with a clutch arrangement when the main rotor is not turning (rotor will be locked during engine starting and engine check-out). The APU will be de-clutched when the rotor is turning. The accessory gearbox will provide power for all hydraulic, electrical, pneumatic, main transmission lubricating systems and all auxiliary mechanical systems. The auxiliary mechanical systems include the rotor brake and the tail rotor. The lubricating system will include all possible main transmission oil system components. The consideration is pre-oiling the transmission by APU prior to main rotor engagement, and to improve oil system behavior during cold-weather starts by pre-heating the lubricating oil.

Fixed controls will be located within the transmission with the hydraulic actuators as close as feasible to the rotor blade pitch horn. Reliability is the consideration.

An acceptable autorotation rate of descent will be provided at the 12-ton transport mission weight. This mission is selected because the 20-ton heavy lift mission specifies external cargo which would be jettisoned during autorotation, while the ferry mission weight is chiefly jettisonable fuel.

The design philosophy is that a selected design approach will be based on comparison of alternate solutions. The following design criteria will apply.

(a) Design torque is taken as the maximum steady-state operating condition. This corresponds to 12000 horsepower at 3600-r.p.m. input.

(b) Limit torque is 200 percent at maximum design torque. No permanent deformation is permissible.

(c) Ultimate torque is 1 1/2 times limit torque or 3 times design torque. Parts may deform permanently. Parts must not fracture.

(d) Assume that there is no flywheel resonance problem.

(e) Cyclic torque components will be present at frequencies corresponding to output r.p.m. and integral multiples thereof. Amplitude of cyclic torque components will probably decrease sharply with frequency, orders above the sixth being usually negligible. It should be assumed that the peak-to-peak wave form will be contained within the envelope of ±10 percent of design torque.

(f) Temperature envelope is -65°F to 180°F. The ship must fly without warmup. Provisions have been made for pre-oiling the transmission prior to start and for some pre-heat of the oil.

(g) Total vibration level shall be ±0.3 g and will occur at integral harmonics of output r.p.m.

(h) Acceleration to speed will be controlled by the free turbine torque-speed curve and the system inertia. A typical free-turbine torque-speed curve may be developed from a smooth curve on a plot of percent torque versus percent speed. The curve connects 100-percent torque at 100-percent speed, 109-percent torque at 91.3-percent speed, 122-percent torque at 80-percent speed, 138-percent torque at 68.5-percent speed and 190- to 200-percent torque at 0 percent speed.

(i) Transient maneuver acceleration shall be -0.5 g (pushover) to +2.5 g (pullout) along the axis of the transmission. Aircraft yaw acceleration is 8 radians per second^2 about the transmission axis. Aircraft pitching acceleration is 3 radians per second^2 about a point (aircraft center of gravity) approximately 8 feet below the turbine plane and on the transmission axis. Aircraft roll acceleration is 12 radians per second^2 about the same point. Crash load accelerations are 20 g's forward, 20 g's downward, and 10 g's lateral, and are the ultimate strength requirement for the transmission mount. The transmission need not be operable after ultimate strength loading.

(j) Operating torque is from 40 percent to 100 percent design torque, with only very short-term transients above 100-percent design torque (assume impacts are not a factor, because of limit torque requirement). Load schedule was not specified for this contract. TBO is 1200 hours. Drive train bearings will be designed to exceed a minimum of 1200-hours B-10 life at full power. It is expected that this specification will result in a prorated B-10 life larger than 2500 hours. The main rotor bearings will be designed to exceed 1400-hours B-10 life, with maximum continuous allowable thrust and hub moment (values used were 80,000 pounds, 1,270,000 pound-inches). The turbine mounting bearings will be designed for continuous operation at the worst operating condition of thrust and moment. This was found to be with two adjacent gas input quadrants inoperative. The loads are 400-pounds thrust, 8770-pounds-inch moment. For the tail

rotor and accessory drive provisions, percentages of total power to be used will be: rotor brake - 20 percent, tail rotor - 7 percent, tail rotor overload - 14 percent, accessories - 2 percent. TBO is also 1200 hours. The load-time schedule will be: 2400 horsepower - 0 hours, 1920 horsepower - 180 hours, 1080 horsepower - 840 hours, 480 horsepower - 180 hours. For the tail rotor drive aft of the accessory box, 1870 horsepower will be used.

Bearing life calculations will use the American Standard Method of Evaluating Load Ratings for Ball and Roller Bearings, published by the American Standards Association on 6 January 1959. (reference 1).

(k) All gearing, shafts, splines, or other mechanical elements which contribute to a major portion of the foreseeable cost of a final production design will be designed for infinite fatigue life at design torque. The stress criteria are: static allowables per ANC-5, shaft endurance limit in bending ±18,000 p.s.i., shot peened ±25,000 p.s.i., tooth bending 40,000 p.s.i. single direction, and ±25,000 p.s.i. reversed, Hertz stress 150,000 p.s.i., scoring 400°F desirable, 600°F allowable (180°F blank temperature). The lubricant is MIL-L-23699. All teeth will be modified to these bending, pitting, and scoring criteria, calculated by the following methods: gear bending stress (reference 2, pp 130-160), gear Hertz stresses - the conventional Hertz equation, and flash temperature - scoring criteria (reference 5, pp 141-142).

POWER TRANSMISSION SYSTEMS FOR HEAVY LIFT HELICOPTERS

A. DESIGN OF POWER TRANSMISSION SYSTEMS

1. Transmission With Integrated Rotor Hub

The inboard profile is shown in Figure 7 and the 3-view of the power plant system is shown in Figure 8. The transmission is designed for 12000 horsepower, 3600-r.p.m. input, 124-r.p.m. output. The overall reduction ratio is 29:1. The chief feature of this transmission is the integrated rotor hub/ring gear.

The integrated rotor hub/ring gear is an annular box section with gear teeth on the inner surface, bearings mounted on the upper and lower surfaces, and lugs for the lead/lag hinges extending from these upper and lower surfaces. A three-piece construction is planned. The member with the internal gear teeth has a central radial flange which extends outward and bisects the box section. Two identical plates bolt to this flange and to smaller local flanges located at the top and bottom of the gear teeth, forming the annular box section of the rotor hub/ring gear. The location of the lead/lag/flapping hinge offset followed from the proportions of the annular box section. The offset is 40 inches, which is 6 2/3 percent of rotor radius. Offset selection for aircraft and center of gravity control is properly a subject of a rotor study, and beyond the scope of this study. The 6 2/3-percent offset is believed to be reasonable.

Five sources of load were used in calculating box deflection to determine mounting bearing deflection. Three of these calculations were by a computer program for frame analysis developed by Kaman for structural analysis of fuselages. Equations were derived to determine the mounting bearing race deflections which the bearing rolling elements must withstand. Deflection for various load distributions may be found in the deflection analysis section of this report.

The combined effect of the various deflections and cycle rates is not obvious. A computer program was modified to include the factors requiring consideration and to make a life determination for this application. The report on

the method of analysis constitutes the appendix to this report.

To reduce local deflections of the rotor hub/ring gear, a large but practical number of blades was selected - seven. To reduce rotor/hub ring gear deflection and gear tooth face width, six planets are planned in the last stage. In this case, consideration was also given to the desirability of using a noninteger relationship between the rotor hub deflection pattern from planet gear loads (six lobed and rotating with respect to the rotor hub) and the rotor hub deflection pattern from blade loads (seven lobed and fixed with respect to the rotor hub). The use of six blades with six planets would superimpose maximum deflection loading simultaneously in six places. It appears wiser to force a staggered superposition. If five or six blades are found to be optimum, the number of planets would be changed to maintain staggered superposition.

The planets are mounted to nonrotating gearbox structure which also provides support for the mounting bearing above the rotor hub. The use of six planets limits the speed reduction of the last stage to approximately 2:1 and, in turn, determines some of the characteristics of the preceding speed reduction stages. Tooth numbers and diametral pitch were chosen tentatively, pending the outcome of the bearing analysis. The preceding speed reduction stages were investigated. The ratio required for the stages preceding the last stage is approximately 14 1/2:1. This is in the range where the split power planetary may be competitive with the epicyclic. Tooth numbers, diametral pitches, and number of planets were selected for a split power planetary and for a two-stage epicyclic speed reduction. The epicyclic appeared lighter and simpler in this instance, and was used.

Bearing B-10 life from the computer program developed for this study was found to be 325000 hours with a rigid race assumption (infinite EI, where EI = stiffness criteria) and 77000 hours with design EI of the annular box. It was felt life could be reduced to effect a weight saving. The following alternatives were available: reduce bearing diameter, reduce bearing cross-section, reduce annular box EI. The transmission stress analysis shows that the transmission constrains bearing diameter reduction because of planet face width considerations and planet bearing size decrease with load increase. Some reduction could be achieved, but the route is not attractive. Bearing

life reduction by bearing cross-section reduction is also not attractive, because the bearing weighs much less than the annular box. The same life reduction achieved by annular box EI reduction would save more weight. This was accomplished by changing box wall thickness to limits compatible with stress requirements. The resulting EI term is 70 percent of the previous value. Bearing life is greater than 15000 hours, B-10. Bearing life should not be reduced to a calculated life of less than 15000 hours, B-10, without experimental verification of load distributions, because of the unconventional nature of the loading.

For the two-stage epicyclic portion, reduction ratio per stage is approximately 3.8:1. At this ratio, it is possible to use six planets/stage with overhung planet mounting. It has been found more desirable to solve the problems associated with six planets with overhung mounts rather than to use a one-piece carrier with straddle-mounted planets, space-limited to four or five planets. The overhung problem is solved by providing structure above the carrier ring to support the trunnion-loading from overhung planets. A design technique has been developed for defining the proportions of this structure, which has been substantiated in several production transmissions over the past eight years.

Design loads uniformly have been kept below those associated with previous successful applications designed for 1000 hours of time between overhauls (TBO). Some of the loading conditions analyzed are seldom critical in conventional transmission work, but must be carefully treated in highly-loaded planetaries. For this reason, each member is discussed in some detail in those areas which have been found critical by previous experience.

(a) Sun - The sun gear proportions must account for a proportionally higher number of stress cycles than the ring and planet gears, and for the rosette hoop stresses applied by radial tooth separating forces. This results in a tooth critical in scoring and a heavy wall thickness requirement. Increased planet tooth thickness, as discussed below, requires a corresponding decrease in sun and ring tooth thickness. In addition, the sun addendum was reduced, thus reducing tooth tip sliding velocity while still maintaining an acceptable contact ratio of 1.40. Break-in technique developed by this Contractor to aid in relieving scoring problems is expected to be applied.

The scoring criterion for these teeth is 425°F which compares with a value of 500°F generally accepted limit, and 600°F used successfully on the UH-2A transmission following developed break-in procedures. Multiple engagements of the sun gear give a large heat transfer problem, which in this case is handled by multiple cooling jets.

(b) Planets - The planet proportions must account for reversed tooth bending loads, centrifugal force, diametrically opposed radial loads from tooth separating forces, and, of course, output torque.

The planet teeth are critical in reversed bending, which has resulted in tooth modification of increased tooth thickness while maintaining standard addendum.

Planet ring-type deflections from high diametrically opposed separating tooth forces can overload the bearings locally, unless sufficient wall thickness is maintained. A wall thickness was selected which limits this local load on a limited number of rollers to a residual load less than the usual proportion of distributed load previously found successful. The fiber stress on the inside surface is the significant criterion for this ring stiffness requirement.

(c) Planet Bearings - The planet bearings have been proportioned for the combined loading case resulting from output torque, centrifugal force, residual diametrically opposed forces from tooth separating loads, and roller end loading resulting from cocking caused by torsional windup and internal bearing clearances. Because a combined loading case must be treated conservatively, B-10 life was established at 1200 hours at full power. Each planet is supported by a single or double row roller bearing with a monolithic cage with in-line pockets. The bearings are cylindrical rollers with end relief.

The planet bearings are mounted on separable inner races on a carrier spindle by ground spacer rings such that clamping the end plates on the inner race results in a total axial clearance of .0004 to .0012 inch for each row of rollers.

The separable inner race sleeve facilitates modification to provide for excess deflections if such are encountered in testing. The modification possible is to bore the internal cylindrical surface at an angle to the outer

cylindrical surface, this angle to be determined by the deflection to be compensated. These inner raceways would then be indexed and, under load, the spindle deflection would be compensated. It is not anticipated that such a requirement would exist, but it is considered desirable by this Contractor to anticipate potential difficulties of this sort in the details of the initial design.

A further advantage of a separable inner race is that when an unacceptable wear limit is reached, this part may be replaced much more inexpensively than could a carrier. This is a worthwhile consideration, since, by geometry, the inner race is the lowest life bearing element.

(d) Ring - The ring gear teeth, by the geometry of internal gearing, are not as highly loaded as sun and planet teeth. They are modified by reduced tooth thickness, however, by the planet tooth increased thickness requirement. An additional modification customarily employed, but not apparently necessary in this case, is to increase addendum to increase contact ratio and decrease localized loading in the back-up hoop. The critical stress is the hoop stress imposed by rosette loading of the ring. It has been found sufficient to attach stiff structural members to the ends of ring gears, such as the conical housings used in this case.

(e) Carrier - The carrier is a statically-loaded ring carrying trunnion loads imposed by each planet. The structure above the ring is proportioned to resist this trunnion loading and can be thought of as combining with the ring to simulate an I-beam in bending. An analysis technique has been developed and substantiated on numerous applications. This allows a confident prediction of spindle deflection under load, which is necessary for bearing internal clearance calculations.

2. Transmission With Segregated Rotor Hub

The inboard profile of this transmission is shown in Figure 9, and the 3-view of the power system is shown in Figure 10. The transmission is also designed for 12000 horsepower, 3600-r.p.m. input, and 124-r.p.m. output. The overall reduction ratio is 29:1.

Tooth numbers, diametral pitch and number of planets were selected for a split-power planetary, for a partial

split-power planetary with one epicyclic stage, and for two different three-stage epicyclic speed reduction arrangements. The three-stage, constant ratio per stage, epicyclic transmission was lightest of all systems considered, and was simpler than either split-power system.

The overall reduction ratio is 29:1. With three stages, the reduction ratio per stage is 3.0+:1. The epicyclic transmission optimizes at this reduction ratio under aircraft transmission design practice. The following parameters are usually found to be in balance at this ratio: planet tooth reverse bending stress, sun gear pitting and scoring criteria, planet bearing capacity, planet bearing spindle stiffness, planet face width to diameter ratio, planet load sharing requirements, ring gear stiffness and strength, and planet wall stiffness and strength. The exceptions to this rule occur usually from centrifugal force loads on the planet bearings at relatively high input speeds. In this application, the input speed is low, and centrifugal force requirements were easily accommodated. The discussion in the previous section on design considerations is applicable here except that eight planets are used in each stage rather than the six planets per stage of that configuration.

The transmission lubricating system was investigated to assure that all requirements can be met. Oil jets are not shown, but are planned for the appropriate locations. Many of the transmission oil system components are to be located in or near the accessory gearbox, including the scavenge pumps, oil tank, pressure pumps, filters, pressure relief valves, cooler blower, and cooler. The accessory gearbox has clutch-connected APU drive with the rotor locked. This arrangement will pre-oil the transmission when the APU is driving prior to main rotor engagement and provide some oil heating for cold weather starts.

Transmission oil cooling requires estimates of losses converted to heat and estimates of the amount of heat rejected by the transmission cases. Both transmissions are subject to roughly equivalent losses; however, this transmission (segregated hub) has, by geometry, smaller case heat rejection. Cooling requirements were calculated by conventional methods for this transmission and are assumed applicable to both, ignoring the small error. The losses converted to heat (16100 BTU/min) less the heat rejected by the transmission cases (3800 BTU/min) equals the cooling load (12300 BTU/min).

Two systems were considered: an unblown convection cooler and a blown cooler. Weights were equivalent, with the blown cooler appearing superior because engine ingestion of heated air can be avoided with greater reliability and also because of the reduced size required for the installation. The cooling system installation is shown on the power plant installation. It is planned to start transmission development with a conservative allowable oil temperature rise of 50°F., although a penalty is paid in the form of the oil pumping requirement (66 gal/min). During development, the allowable temperature rise of the cooling oil in the gearbox can be investigated under actual operating conditions. Hopefully, a higher allowable temperature rise can be demonstrated and pump capacity reduced. It is calculated that a 15- to 20-horsepower blower is sufficient, with cooler dimensions of 765 square inches of radiator core of 3.75- to 4.00-inch thickness. A ram air intake is planned to allow blower disconnect at 60-knots forward speed, sea level standard. The details of the ram intake require experimental determination and were not explored. The oil system requires 24 gallons of oil. The system weight is calculated to be 235 pounds. One of the advantages of the TIGR arrangement is that the transmission oil system is greatly reduced in size and complexity compared to oil system requirements for the usual multiple engine/transmission practice, not only saving weight but also greatly improving reliability. Reliability is also enhanced by the planned ability to survive blower disconnect, whether deliberate or inadvertent, at the relatively low forward speed of 60 knots.

The rotor hub is integral with the rotor mast. This may increase manufacturing cost, but greatly reduces weight. Any joint between the hub and the mast is exposed to lift, maneuver, and crash loads, must transmit torque with steady and cyclic components, and experiences rotating bending loads. It is estimated that this joint adds 20 to 30 percent to hub weight. Because increased weight will increase fuel consumption and an extra joint will reduce reliability and require more maintenance, there is a trade-off between manufacturing costs and operating costs. Because trade-offs of this nature are nearly always dominated by operating costs, the integral hub/mast was selected as most efficient.

The hub uses seven blades with 40-inch offset to the lead/lag/flapping hinge axes, identical to the integrated rotor hub/ring gear. A weight saving could be shown by reduction in number of blades and offset, because this configuration

does not have a transmission or mounting bearing constraint. It was decided that reduction in number of blades and offset, although mechanically feasible, would require substantiation by rotor dynamic analysis and by analysis of aircraft center of gravity and maneuver control parameters. These analyses are beyond the scope of this transmission study. Further, change in these parameters without substantiation would introduce a complicating factor in any comparison or evaluation of the two transmission approaches. Sufficient data were assembled to show that five and six blades are within the range of acceptable performance and structural consideration. The rotor design and analysis program planned by USAAML is expected to optimize the number of blades and the offsets. The seven-blade and 40-inch choices appear to approach the high limit in both cases.

Hub design uses a "framing" feature that greatly reduces the stress on the cantilever arms that extend radially outward from the central spindle to the lugs for the lead-lag pin. The arms are clamped by the pin to the inner races of the lead-lag bearings and to the cross, and are forced to act as an open-celled frame with moment reaction capability at all four corners. The method of analysis is by a variation on the methods presented by Kleinlogel (reference 7, pages 151-153). The arms are open U-channels in cross-section. The depth increases toward the spindle until the two U-channels merge into an H-beam with continuous flanges. The central spindle is stiffened by a continuous circular flange which forms the cross bar of the H-section and the attachment for the lead-lag damper between the beams. The hub has been stress-analyzed for loads, but further refinement is believed possible. This requires the use of techniques such as photoelastic investigation.

3. Related Areas Affected By Transmission

(a) Blade Retention, Articulation and Control - The lead/lag hinge and flapping hinge are coincident in this design. Rotor blade forces and moments were calculated, the cross proportioned and a wire bundle retention designed. Two approaches to blade feathering articulation were considered in keeping with the design approach to this study program, which provides for comparison of alternate solutions. In the first, the blade spar forms the internal spindle supporting feathering bearings within a sleeve which terminates in a clevis at the flapping bearing axis. In the second, the blade spar forms the

outer sleeve and the internal spindle is integral with the clevis at the flapping bearing. The second is the conventional approach. Both have advantages and disadvantages peculiar to the approach. On balance, the conventional approach appears superior. To decrease the massiveness of the spar extrusion, the outer spindle was made a separable piece, attached to the spar by a two-bolt system, and the material changed to titanium. The inner spindle and feathering bearings were proportioned for starting inertia loads. The inner spindle and cross are cut from one piece of material - thus eliminating a troublesome joint in a critical area. The trapped cross concept was found to be feasible in this size. The static droop stop with a centrifugally operated flapping stop are based on a successful design now in operation. For lead-lag damper design, ground resonance avoidance usually requires greater capacity than does rotor flight dynamics. In this case, the damping capacity is more than sufficient for flight and appears satisfactory for the ground resonance criteria for normal inertia and landing gear relationships. Any changes of damper capacity required by later ground resonance analysis can be accommodated easily. The pitch horn is splined to the outer spindle and clamped with a nut. Pitch horn loads were derived by parametric analysis.

(b) Fixed and Rotating Controls - Three control systems were investigated. The first was the conventional approach, that is, swashplate external to the transmission and controlled by three hydraulic actuators, with links to the blade pitch horns located on the flapping bearing axis. The second used a small swashplate above the rotor hub with linkage controlling seven hydraulic actuators mounted on the hub and controlling each of seven blades through pitch horns located on the lead/lag bearing axis. The third approach, resulting from comparing these two solutions and attempting to achieve the advantages of each, is three hydraulic actuators controlling a small swashplate above the rotor hub with linkage to the pitch horns on each blade.

The first approach was proportioned to obtain a basis for evaluation. The second approach was designed in some detail. Pitch horn location on the lead/lag bearing axis was investigated for stability with respect to flapping-pitch coupling and lead/lag-pitch coupling. Stable and unstable regimes were identified and considered. This appears to be as satisfactory a location of the pitch horn as the flapping bearing axis and, in some respects, is an improvement. The pitch horn loads were determined

by parametric analysis, and the power required to vary blade pitch cyclically was determined.

The hydraulic actuators are inefficient motors - seal drag and flow losses penalized the individual actuator system with a power penalty, as they must complete a cyclic excursion with each revolution of the rotor. The power penalty was converted to equivalent weight. The individual actuator system was found to be superior to the conventional external swashplate system because of the much higher weight of the conventional system. Power penalties are not attractive, and a solution was found where three actuators remain in the nonrotating field mounted within the rotor hub on nonrotating structure, thus avoiding the power penalty, and the swashplate is reduced in size by relocation above the rotor hub. The third solution was designed for both configurations with small modifications required between the two concepts. The signal from the swashplate to the blade pitch horn is transmitted by a torque tube on each arm. The torque tube on each arm is a feature of interest to both designs. It is planned to modify control torsional spring rate in conjunction with blade torsional spring rate to achieve controlled higher harmonic feathering. The torque tube is the structure selected for potential modification. The illustrated torque tubes are, therefore, to some extent pictorial rather than precise. The controls are shown on the inboard profiles and 3-view for both concepts (See Figures 7, 8, 9, and 10).

(c) Affected Portions of Hydraulic Systems - Hydraulic power needs have been identified in two areas: for primary flight controls, and for gas generator starting. The HLH controls have been designed to meet the requirements as stated in MIL-F-18372 (Aer). The controls are Type III - power operated flight controls (1.2.1). They meet the requirements for dual power control systems (3.1.2.3.2) wherein two completely independent power systems are used, dual system failure (3.1.2.3.2.1) wherein one unfailed system can meet the performance requirements of the aircraft, and power control override provisions (3.1.2.6), wherein direct pilot effort is applied to a frozen or jammed valve.

The hydraulic controls are shown in the inboard profiles (Figures 7 and 9). Three actuators control the location of the swashplate and are mounted on nonrotating structure within the rotor hub. Tandem cylinders in the same housing are acceptable, per 3.1.2.3.2. The hydraulic manifold

is centrally located within the rotor mast. One system is connected to the upper cylinders and one system to the lower cylinders. The valves are actuated by push-pull rods within the rotor mast. The actuators are of the following type - valve motion causes actuator motion in the same direction - which reduces valve travel to a limit set by response speed. Two separate hydraulic pumps are located on the accessory gearbox, with two accumulators located in the nacelle compartment. This pump location allows controls check-out by APU power while the rotor is locked and, because the rotor overrides the APU, provides control power while the rotor is turning. The pumps and accumulators are shown in Figure 6, the power plant installation.

Also shown is the provision for the other hydraulic system requirement - gas generator starting. Hydraulic or pneumatic starting is required for the T64, and must be APU powered. A separate hydraulic starting motor is provided on each gas generator with a selector valve directing power individually. Again, the APU/rotor override feature will allow ground or airborne starting. The starting power requirement sizes one pump (50 horsepower) and accumulator. The other pump is estimated to be 25 horsepower.

(d) Affected Portions of Electrical Systems - Electrical power needs have been identified for four purposes: the beacon light (MS 25277-1), the torquemeter/transmission analyzer, rotor deicing, and instrumentation. In addition to the usual required monitoring and warning light instrumentation, a rotor overspeed recorder is included.

It has long been recognized that the ideal location for the beacon light is above the rotor hub. With nonrotating structure within the hub, this is entirely feasible, as is shown on the inboard profiles and the 3-views (Figures 7, 8, 9, and 10).

The torquemeter/transmission analyzer and the rotor overspeed recorder will be described more completely under Maintainability (page 62). The electrical provision requirements for the torquemeter/transmission analyzer are for a recorder, an integrator, and a multidrum transient-indicator counter. The rotor overspeed recorder is a small, solid-state, production device to the located in the cockpit with a visual signal for each desired overspeed range, thus taking the guesswork out of power system maintenance and realiability requirements in the event of an

overspeed occurrence.

Rotor anti-icing was studied qualitatively and quantitatively, although a requirement for anti-icing was not specified in this contract. The duration of the ferry mission is over thirteen hours, leading to the conclusion that fluid anti-icing techniques lack sufficient endurance for this mission. Electrical anti-icing meets the endurance requirement and was therefore investigated. The slip rings and brush assembly would be located around the mount for the navigation light as is shown on the inboard profiles (See Figures 7 and 9). The small radius with low speed promise long brush life, while the location is unusually accessible. The blade distribution spider is also the support for an aerodynamic fairing. The quantitative values, based on number of blades, blade contour, radius, chord and velocity, may change significantly with the rotor study. Analysis for approximate power requirements for the rotor used in this study show 30 KVA required. Refined design, possibly supported by some experimental research, may lower power requirements to 20 KVA. The anti-icing generator is located on the accessory gearbox and has the same benefits as the hydraulic pump location from the APU/rotor-override accessory system.

(e) Affected Portions of Aircraft Structure - A dynafocal transmission mount was first considered, but proved not compatible with engine and transmission geometry. Next considered was the development of a single main transverse structural frame to which the transmission would be shear-mounted with tension and compression struts required. This, in turn, gave way to the final configuration: flanged horizontal transmission mounts on either side of the transmission which are integral with the transmission upper case containing the main rotor lift bearings. The mount-beams attached to these flanges are terminals of the main transverse structural element of the HLH. A tapered, triple-frame, double-celled structure, which is attached to the transmission case, closes one cell at the HLH roof and contains another cell which is the cargo compartment. The frames are on 30-inch centers at the roof line, 19-inch centers at the transmission mount. The cargo hook for sling loads is attached to this member as is the main landing gear. The cargo compartment deck is cantilevered in both directions from this member, also the roof structure. Thus, landing loads, sling loads, cargo loads and rotor loads all have the same primary path for greatest structural efficiency.

A nacelle is planned which encloses the gas generators, engine and transmission, and the accessories. The nacelle is shown in Figure 6 , the power plant installation. The forward section of the nacelle contains louvers for induction of cooling air which is drawn through the nacelle by exhaust ejection. The nacelle has latched parting planes between each gas generator to allow removal of individual engine covers or, in some environments, removal of the entire forward nacelle.

The after half of the nacelle is also removable, sliding on rails provided on the roof structure. The roof within the nacelle is a work platform 12 feet wide by 28 feet long.

The main transverse triple frame has permanent steps fore and aft, port and starboard, that lead to semi-permanent work platforms located above the turbine and both fore and aft of the main transverse structure. For a standing man, rotor system components are at work bench height from these platforms, as is shown in Figure 6 , power plant installation. The aft platform is removed for power system installation and removal.

An HLH design advantage is apparent in that changes in tail boom weight during design may be accommodated easily by fore or aft change in gas generator design location. The 2500-pound gas generator package may be moved small distances without penalty to the power transmission system (small change in duct length).

(f) <u>Accessory Provision</u> - Accessories are mounted on an accessory gearbox, shown on the power plant installation (See Figure 6). The gearbox is driven by a drive shaft, which in turn is driven by a spiral bevel gear set powered by the turbine. The turbine is driven by the gas generators or, in the event of a power failure, by the main rotor, with the speed-reducing gearbox acting as a speed-up gearbox. The accessory gearbox is also driven by an APU with a clutch arrangement to permit accessory drive with the rotor locked. The accessory gearbox provides power for all hydraulic, electrical, pneumatic and auxiliary mechanical systems. The auxiliary mechanical systems include the rotor brake and the tail rotor. The accessory gearbox contains some main transmission oil system components, including the scavenge pumps, oil tank, cooler, cooler blower, oil pressure pumps, filters and pressure relief valves. This arrangement will pre-oil the transmission upon starting the APU prior to

rotor engagement, and will improve oil system behavior during cold weather starts by pre-heating the oil.

(g) Tail Rotor Drive By Hypercritical Shafting - Aircraft drive shafts have been designed traditionally to operate at speeds below critical speed. This requirement controls the relationship between shaft speed, cross-section, material, and the distance between shaft supports. If shafting can be operated at speeds above the critical speed, a weight saving will result. Shaft speed can be increased, thus lowering transmitted torque for a given power, shaft cross-section can be reduced, and shaft supports can be eliminated or modified. The Army has funded research programs in this area and investigation is continuing. The HLH tail rotor drive appears to be a suitable application and was investigated.

The weight was calculated for steel tail rotor drive shafts at three drive shaft speeds - 3600 r.p.m.; 7200 r.p.m.; and 10800 r.p.m. based on presently available research data. The power is 1870 horsepower. The weights are 203 pounds, 155 pounds and 109 pounds, respectively. Consideration given to the trade-off factors involved in using hypercritical shafting for the HLH comparison study indicate that the weight may be used for an aircraft projected for the 1967-1972 time frame, although additional development work is required prior to that time. It is recognized that test programs now in process may be expected to confirm or discourage hypercritical shafting as a practical expectation. The parametric analysis used to determine HLH design gross weight does not use hypercritical shafting because it is not yet proven.

(h) Power Plant and Performance - TIGR is an extremely compact power transmission system composed of lift-fan type turbine, transmission and rotor hub, requiring close coordination between engine and transmission design for maximum design efficiency. This transmission study was supported continuously by detailed engine information, as discussed in the following nine areas.

(1) Weight and Center of Gravity Location - In estimating these weights, reference was made, where possible, to similar parts in existing engines such as the T64 fourth-stage bucket root and the X-353 inlet scroll. Stress was generally maintained at levels at least as low as current practice with commonly-used materials. Bucket stresses were adjusted downward from current levels due

to the impact loadings resulting from partial entry. As the design at this point in time is conceptual in nature, an estimate of the possible error is included in the comparison. Engine oil cooling by a fuel/oil cooler is projected for another program and is considered in this study. The weight summary is shown in Table I below.

It was found that separate exhaust diffusers for each quadrant were advantageous. The exhaust pipes are located as best fits the installation, two to port and two to starboard. This defined the exit angle with respect to the aircraft, affecting weight and center of gravity of these items. Also, plans to locate rotor controls within the rotor mast require the turbine hub to be larger in diameter than would otherwise be required. The rotor moment of inertia (titanium) is 443 pound inches/second^2. The center of gravity location (based on steel disc) is 73.7 inches forward from turbine shaft centerline, 14.7 inches below central plane of turbine rotor, and 0.4 inches to port from aircraft centerline.

TABLE I

MULTIPLE GAS GENERATOR/REMOTE TURBINE WEIGHT SUMMARY

	Weight (Pounds)	Possible Error
4 T64/S4A Gas Generators With Fuel/ Oil Coolers	2440	0
4 Shutoff Valves (actuator weights included in Table IX as miscellaneous weight)	66	$\pm$ 13
Hot Gas Ducting (16 feet)	58	$\pm$ 12
Inlet Scroll Assembly	230	$\pm$ 33
Turbine Rotor	252	$\pm$ 25
8 Bellows Expansion Joints	64	$\pm$ 13
Exhaust Diffuser and Collector	212	$\pm$ 40
Support Structure and Seals	141	$\pm$ 28
Total Propulsion System Weight	3463	$\pm$164

(2) Turbine Power and Performance - The following tabulation presents the minimum performance of the complete turbomachinery system for zero flight speed at 3600-power turbine r.p.m., with idle bucket windage losses reduced 60 percent by means of a retractable wall behind the turbine buckets.

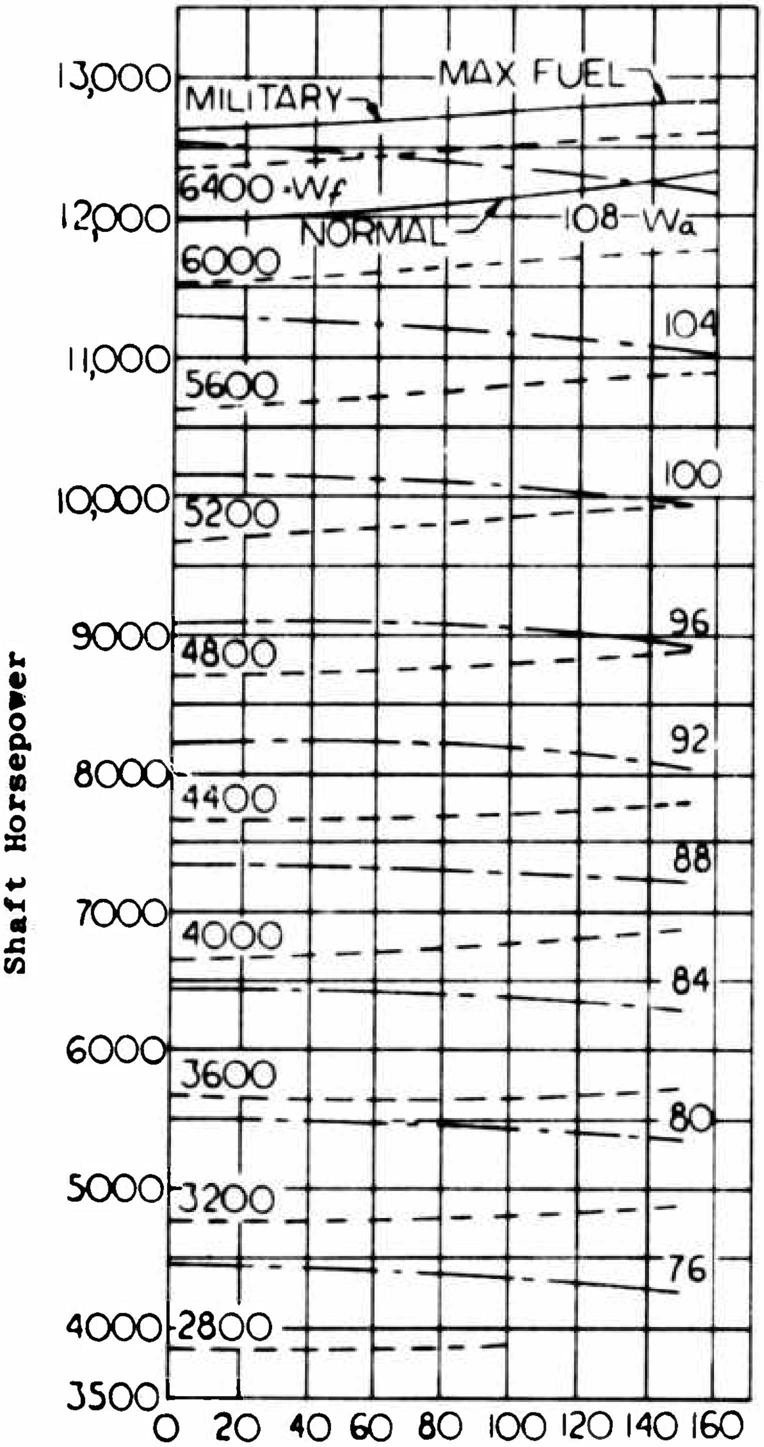
TABLE II

MULTIPLE GAS GENERATOR/REMOTE TURBINE PERFORMANCE

Altitude	T_2 ^oF	SHP	SFC	Setting	Gas Generators Out
0	59	12630	.519	MIL	0
		11410	.522	---	0
		9050	.545	---	0
		6600	.604	---	0
		4440	.692	---	0
		-738	----	Auto. Rota.	4
6000	95	9140	.534	MIL	0
		8245	.544	---	0
		7380	.554	---	0
		6430	.575	---	0
		6680	.549	MIL	1
		4235	.576	MIL	2*
		4279	.570	MIL	2**
		1860	.656	MIL	3
		-552	----	Auto. Rota.	4

* Opposite Quadrants
** Adjacent Quadrants

(3) Airflow, Fuel Flow and Shaft Horsepower - Airflow, fuel flow and shaft horsepower at various flight speeds is shown in Figure 4 below. All of the performance shown herein is without regeneration. Other performance data are presented in section C, page 70.



SEA LEVEL

Preliminary Estimated Minimum Performance For 3600 R.P.M. TIGR Turbine With 4 T64/S4A Gas Generators

T64/S4A Data Were Used To Scale TIGR Turbine Performance From Sea Level Static, Standard Day

Figure 4. Airflow, Fuel Flow, And Shaft Horsepower Versus Flight Speed

(4) Regenerator Installation Considerations - Regeneration was investigated in a general manner without detailed treatment, because it is not apparent that regeneration is appropriate to the HLH missions. Regeneration pays for its weight and complexity only through extended operation at part power settings. The ferry mission is the only HLH mission that approaches extended operation. However, the HLH is a multi-engine system with provisions for removing surplus engines to maintain high power settings on the remaining operating engines.

If regeneration were to be added to the TIGR system, there are a number of practical considerations which would suggest that the regenerators should be placed upstream of the power turbine instead of downstream as in a conventional regenerative system. The considerations include simplicity of high and low pressure ducting, avoidance of back pressure on the large diameter power turbine seals, and avoidance of the difficulty of keeping the gas streams separate and without interactions through interconnecting passages in the turbine that could cause control complications. Mechanically, the installation of the regenerator just behind the gas generator would be relatively straightforward. The high gas side pressures at this location would be conducive to a reduction in size and weight. The mechanical advantage of increased density would be partially mitigated by the fact that gas side temperatures are approximately 300° higher.

Before any extensive effort is devoted to regeneration for HLH systems, a cost effectiveness study would be required to demonstrate that such effort would be worthwhile. If regeneration is shown to be desirable, the flexibility of the TIGR concept permits an attachable regenerator to be developed for use only on appropriate missions such as the ferry mission.

(5) Power Required to Motor the Turbine - The chief contribution to motoring power is from idle bucket windage. This windage can be minimized by providing a close-fitting and smooth tunnel for the bucket passage. This technique is used for the inactive arc of the X-353 lift fan and in most steam turbines. The nozzle diaphragm partitions provide a fairly smooth tunnel wall on one side of the bucket, because of their small angle relative to the plane of rotation. Pivotable vanes will be provided on the other side of the bucket. The pivotable

vane arrangement is similar to the variable compressor inlet stator arrangement used to achieve desirable engine starting characteristics. During normal operation, these pivotable vanes will be faired into the flow stream. When the arc is inactive, the vanes will be pivoted to a closed position, forming a smooth wall behind the turbine. Large steam turbine data indicates that such a smooth wall 1.5 inches from the buckets would result in approximately one-half the windage losses. The minimum practical distance between bucket and wall is believed to be approximately 0.8 inches, and at this spacing the idle bucket windage would be reduced to about 40 percent of the unblocked value. In the absence of test data substantiating a lower figure for motoring power, this value was used for calculating autorotative rate of descent.

The pivoted vanes are the free-wheeling provisions for the engine. Comparison with conventional drive-train, free-wheeling units is in order. The pivoted-vane is static hardware. The conventional is dynamic. Pivoted-vane failure is noncatastrophic, failure of conventional free-wheeling units is catastrophic. The failure rate of the pivoted vane is less than .00001 failures per flight hour for the four quadrant system required here. Fully developed sprag clutches, after exhaustive development effort, have a failure rate of .00032 failures per flight hour for the four units which would be required here.

(6) Heat Rejection - Section VI of the T64 Installation Manual, SEI-123, provides complete data on allowable skin temperatures and bay cooling requirements. This is sufficient to show the heat rejection characteristics of the T64/S4A gas generators. The ducting and scrolls of the TIGR turbine system will tend to stay at the gas temperature. Upstream of the turbine rotor, the gas temperature at military power does not exceed 1270°F with the T64/S4A or 1475°F with the T64/S5A. Downstream of the turbine rotor, the gas temperature does not exceed 940°F with the T64/S4A or 1095°F with the T64/S5A. The design of the TIGR turbine includes provision for insuring that the turbine rotor is adequately and evenly cooled on both sides provided that the compartment air in the cavity below the turbine does not exceed approximately 500°F.

An ejection cooler is planned for the entire nacelle. Each engine exhaust terminates in a 22-inch-diameter pipe leading into a short ejector. Based on a 940°F exhaust gas temperature, the exhaust velocity at military power is

330 feet per second (FPS) with 27-pounds-per-second airflow. Test of an exhaust ejector on the T-60 turbine gave an induced flow of 15 percent of motive flow with an exhaust velocity of 290 FPS. Assuming an induced flow of 10 percent of motive flow, any one T64/S4A can draw a minimum of 160 pounds per minute (2100 cubic feet per minute). A total of 640 pounds per minute (8400 cubic feet per minute) of cooling airflow is available with all four gas generators operating, with a potential of removing at least 25000 BTU/minute for a 200°F allowable temperature rise. Less than this amount will be used following an analysis which includes insulation/air blanket trade-off study.

(7) Turbomachinery Deflection and Dynamics - With respect to critical speed, analysis, including the centrifugal stiffening effect of the rotor, indicated that less than a 2-inch shaft diameter was required in the case of a cantilever shaft restrained six inches from the disc center to zero displacement and zero angular deflection, and that shaft size of 5.4 inches (approximately the same as X-353-5 with a similar wheel weight) would be adequate if the first bearing were fairly close to the wheel and the shaft were equally stiff between bearings. The TIGR installation requirements automatically exceed these minimums, because of the internal control rod location. The turbine/hub design includes a multiple piston-ring seal and slinger, as a safety feature, in the event of leakage at the lower gearbox seal.

With regard to lateral vibration of the rotor, there are two modes to be considered. One is the lateral vibration in which the periphery bends into a sine wave with 4, 6, 8, or 10 nodes designated as the flexural modes of n = 2, 3, 4, and 5 respectively. The first two of these flexural modes were important in the design of the X-353-5 lift fan rotor. In this design, however, with the double cone disc presenting a much higher spring rate against lateral deflection, this type of vibration is not expected to be a problem. The designer is free to choose the cone angle of the disc with little effect on disc weight, and this provides a potent design control for insuring that such vibrations occur well above the 3600-r.p.m. speed of the rotor.

The other mode of vibration might be called "panel" vibration of the large thin disc sections. Again, stiffness against such vibrations is a strong function of the curvature of the panel, which, in turn, depends upon the

cone angle of the disc. In addition, the quoted weight for the rotor contains an allowance for local stiffening flanges which could be used to raise the frequency of local sections above the operating speed range or out of resonance with excitations transmitted from the gearbox.

The mounting of the static parts of the assembly provides adequate compensation for thermal growth. The mounting cone for the scroll assembly has cutout slots that allow the difference in temperature between the inlet scroll and gearbox to be accommodated by low-stress bending of the remaining curved panels. These slots are also provided to allow the cooling flow from the upper disc surface, together with any hot gas leakage from the upper seal to be vented into the open. The disc windage between the disc cover and the upper surface of the disc provides the small amount of pumping required to insure positive and continuous cooling airflow.

The mounting cone also transmits all of the inertia, reactive and pneumatic loads from the static structure into the gearbox with one exception. The 45-degree elbows with expansion joints at each end, provided for relief of thermal expansion and inertia loads between the engines and the scrolls, have an unbalanced piston force which must be resisted by aircraft structure.

The following loadings were estimated for the planned configuration with T64/S5A gas generators. These loads do not include inertia forces resulting from aircraft maneuvers. It is assumed that the flight maneuver loads would be in compliance with MIL-E-8593(ASG) (reference 9, paragraph 3.14).

(8) Loadings (includes T64/S5A loadings where they are more severe than the T64/S4A).

Rotor Axial Thrust	=	600 pounds forward to 800 pounds aft*
Torque Applied to Gearbox by Static Parts	=	310000 pound-inches in plane of rotor opposite to rotor torque
Pitching Moment Applied to Gearbox by Static Parts	=	-538000 pound-inches if mounting flange is 11.4 inches above center of turbine wheel
Ducting Elbow Forces	=	5660 pounds at bisector of each 45-degree elbow
Horizontal Force Applied to Gearbox by Static Parts	=	21000 pounds toward rear of aircraft
Vertical Force Applied to Gearbox by Static Parts	=	21000 pounds upward

*Four gas generators are used. Each supplies gas to one inlet quadrant of the turbine. The turbine mounting bearings must withstand continuous operation, with all gas generators operating (thrust only), and with one, two, or three gas generators not operating (thrust plus moment). All combinations were analyzed. The most highly loaded case occurred with two adjacent quadrants inactive - this case was used for bearing life calculation. Static load survival included turbine precessional force and inertia force from aircraft pitch and roll acceleration per MIL-E-8593 (ASG) (reference 9, paragraph 3.14). This is described more fully in the section on analysis of power transmission systems - Dynamic Behavior.

(9) <u>Producibility of the Gas Generator-Turbine Portion of the TIGR System</u> - The propulsion system consists of four standard T64-GE-12 (T64/S4A) gas generators, interconnecting ducts, and the remote power turbine.

The 3400-SHP model of the T64, which until recently has been called the T64/S4A, now has been officially designated T64-GE-12. Funding has been released by BuWeps for calendar year 1965 "go-ahead". Compressor testing is in process and component activity has been initiated in the turbine and combustor areas. Qualification is planned for late 1967. It is anticipated that YT64-12 engines can be made available commencing in December 1966, and T64-12 engines in the first quarter of 1968. Since the T64-12 is a growth version of the T64-6, the risk of advance release is low; hence, engines or gas generators to the production parts list can be made available prior to engine specification.

The above dates can be improved upon if the funding rate can be tightened. The T64-12 engine can be qualified in thirty months from 1 January 1965, at no increase in development program costs, provided funding is available at the required rate by 1 April 1965. On this basis, the PFRT could be completed in September 1966, followed by qualification for production in June 1967. Ground test engines could be provided prior to September 1966, if required.

All engine models require a production lead time of 12 months. It is expected that the T64/S5A will be developed as a derivative of the T64-12.

The power turbine design employed in this propulsion system is entirely based upon proven technology. For instance, the turbine buckets are solid castings having an aerodynamic shape identical to the X-353-5 lift fan buckets. The chords are shorter than the X-353-5 buckets, but the number is increased to produce the same aerodynamic solidity. The bucket shanks and dovetails are identical to those of the fourth stage T64 buckets. The double cone disc selected for this design is larger in diameter than the double cone discs used in the X-376 pitch fan (J35/LF2 lift fan) or the CF700 fan rotor and has a thinner web section. However, other engines have provided experience in the manufacture of large thin discs which is applicable. Forgings of the required size can be obtained in either steel or titanium.

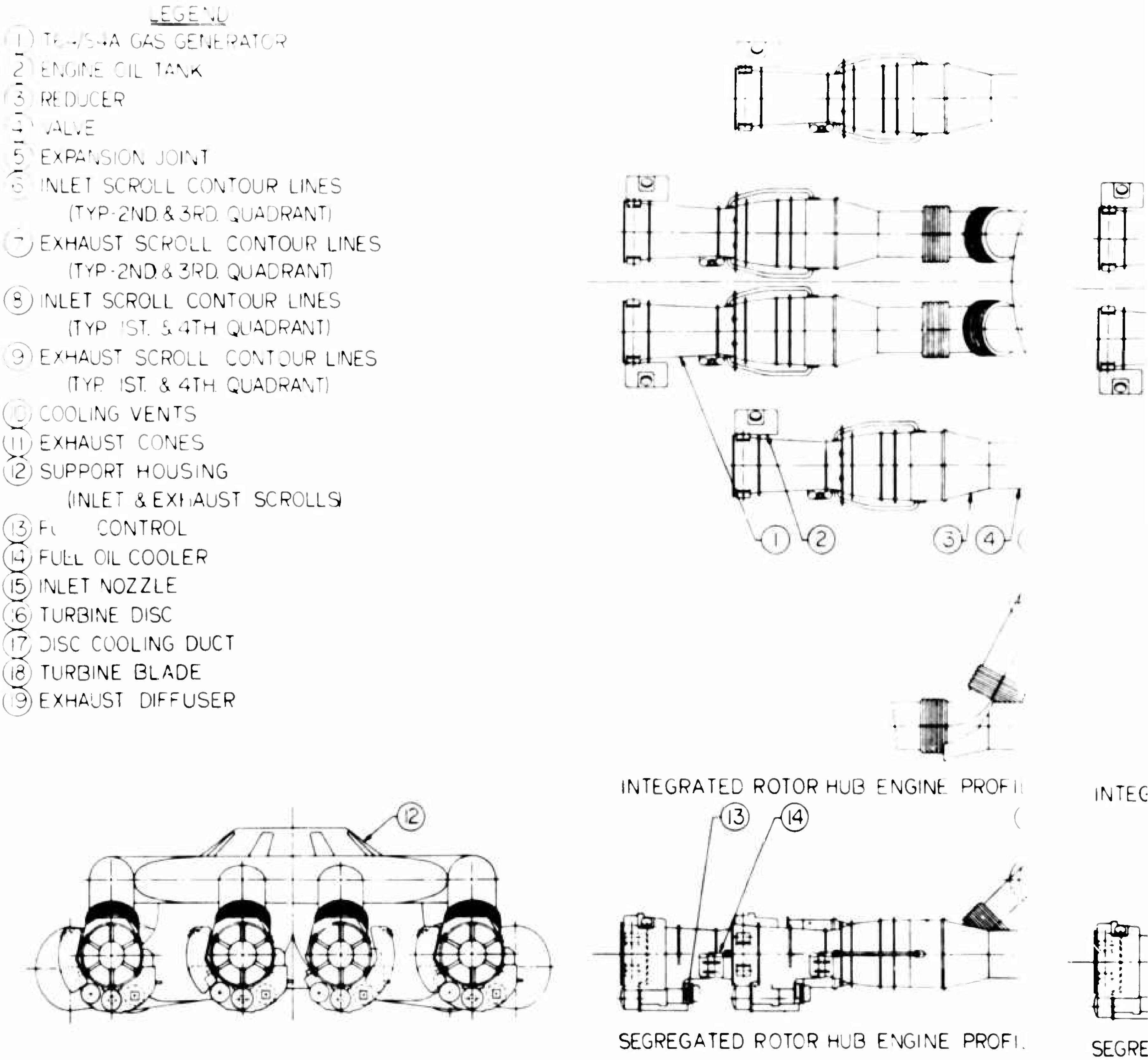


Figure 5. Engine

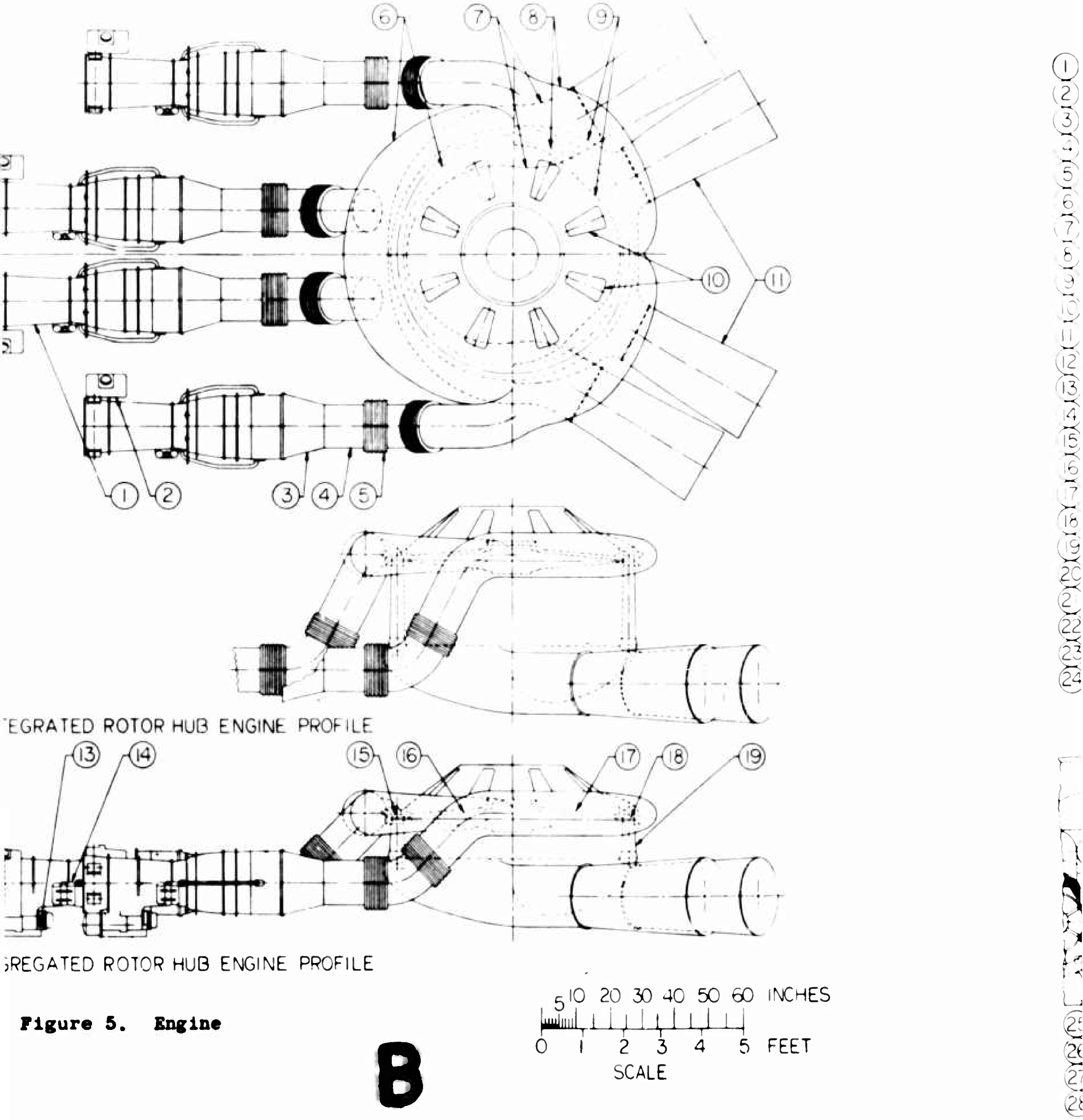
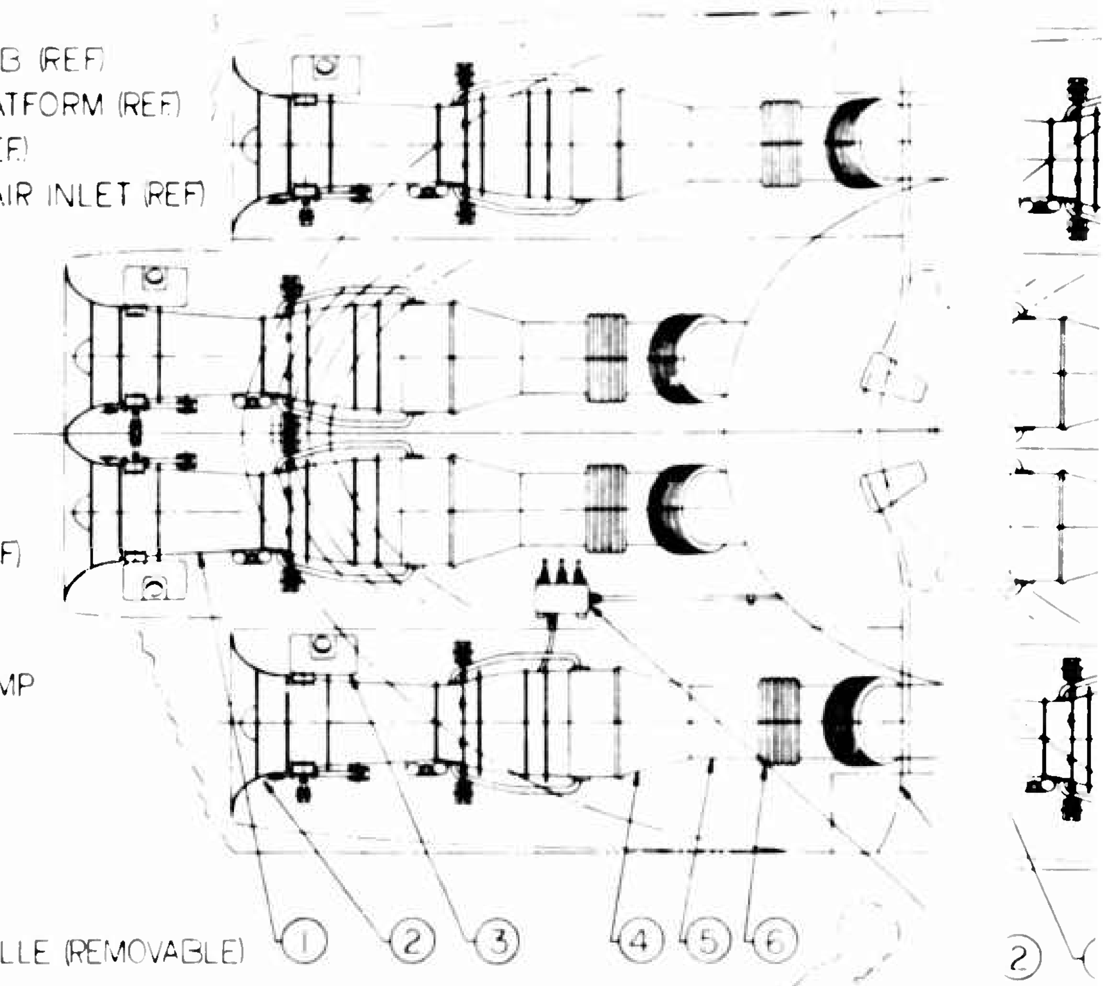


Figure 5. Engine

B

LEGEND

1 T64/S4A GAS GENERATOR
2 AIR INLET
3 ENGINE OIL TANK (REF)
4 REDUCER (REF)
5 VALVE (REF)
6 EXPANSION JOINT (REF)
7 HYDRAULIC STARTER SELECTOR CONTROL
8 NACELLE PARTING LINE (REF)
9 DRIVE SHAFT
10 ALIGNMENT COUPLING
11 ROTOR BRAKE
12 NACELLE INTEGRATED HEAT EJECTOR (REF)
13 GENERATORS
14 FLIGHT CONTROL HYDRAULIC PUMP
15 FLIGHT CONTROL & STARTER HYDRAULIC PUMP
16 AUXILIARY POWER UNIT
17 NACELLE REMOVAL TRACKS (REF)
18 TRANSMISSION OIL TANK
19 HYDRAULIC OIL ACCUMULATORS
20 FWD ENGINE MOUNTS
21 AFT ENGINE MOUNTS
22 COOLING LOUVERS (REF)
23 FUEL CONTROL (REF)
24 STARTER
25 DUCT MOUNTS
26 FWD NACELLE (REMOVABLE PANELS)
27 TRANSMISSION SUPPORT STRUCTURE (REF)
28 TRANSMISSION
29 ROTOR HUB (REF)
30 WORK PLATFORM (REF)
31 STEPS (REF)
32 BLOWER AIR INLET (REF)
33 AFT NACELLE (REMOVABLE)
34 BLOWER
35 PLENUM CHAMBER (REF)
36 COOLER
37 CABIN ROOF (REF)



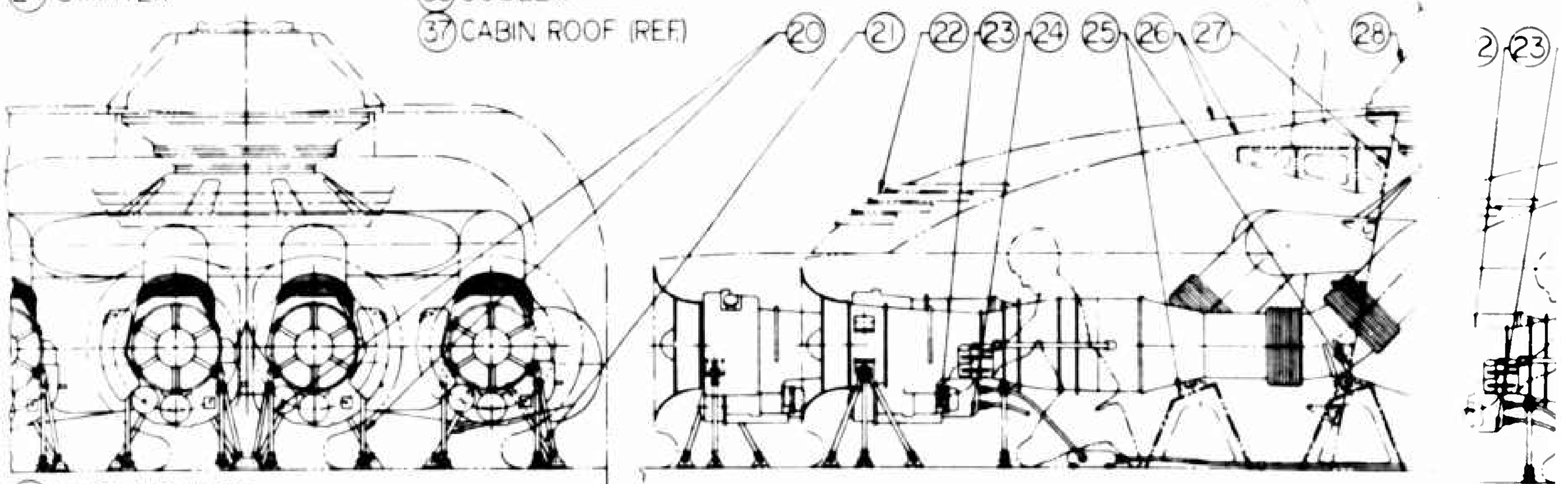
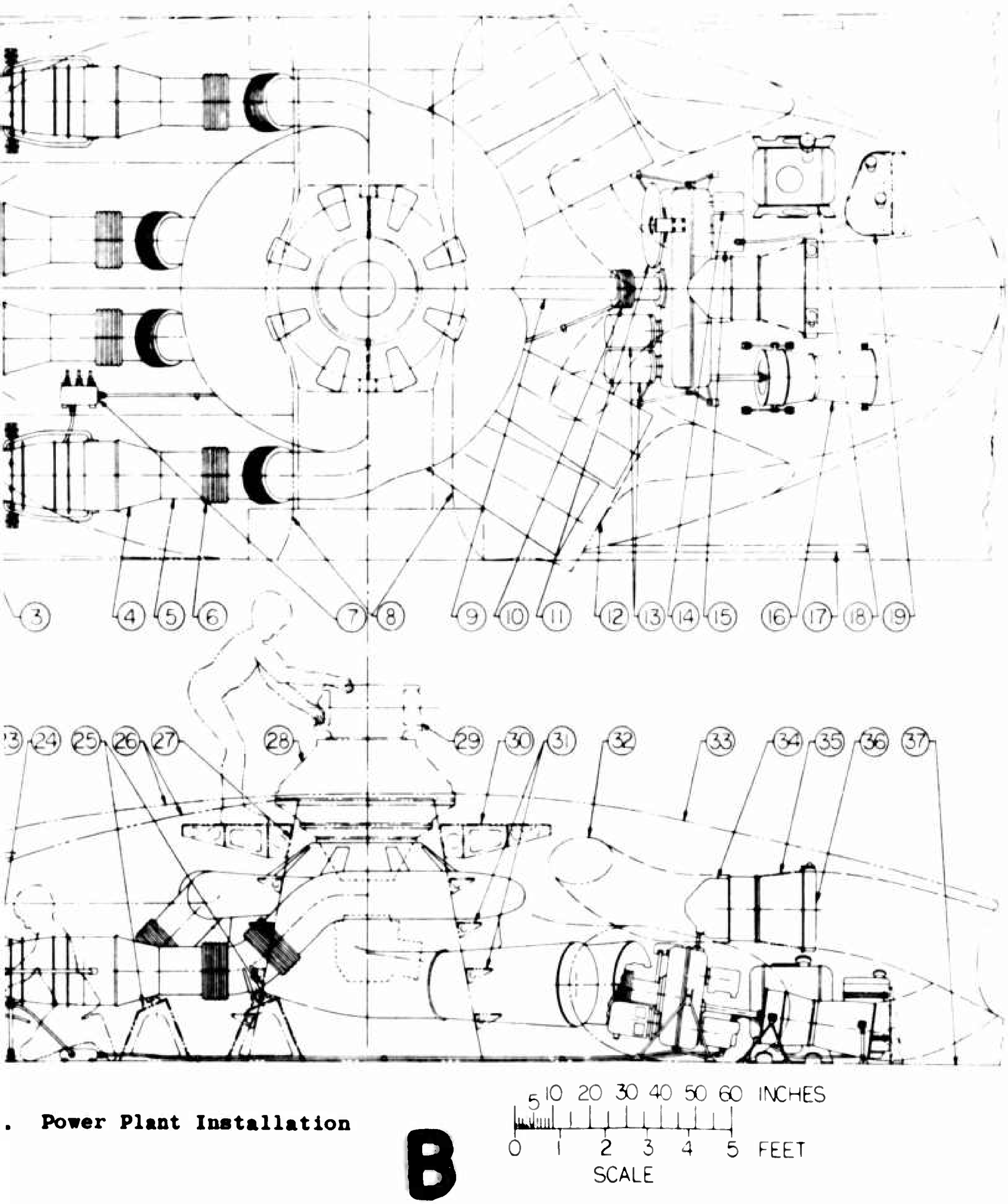


Figure 6. Power Plant Installat. e 6.

A



. **Power Plant Installation**

B

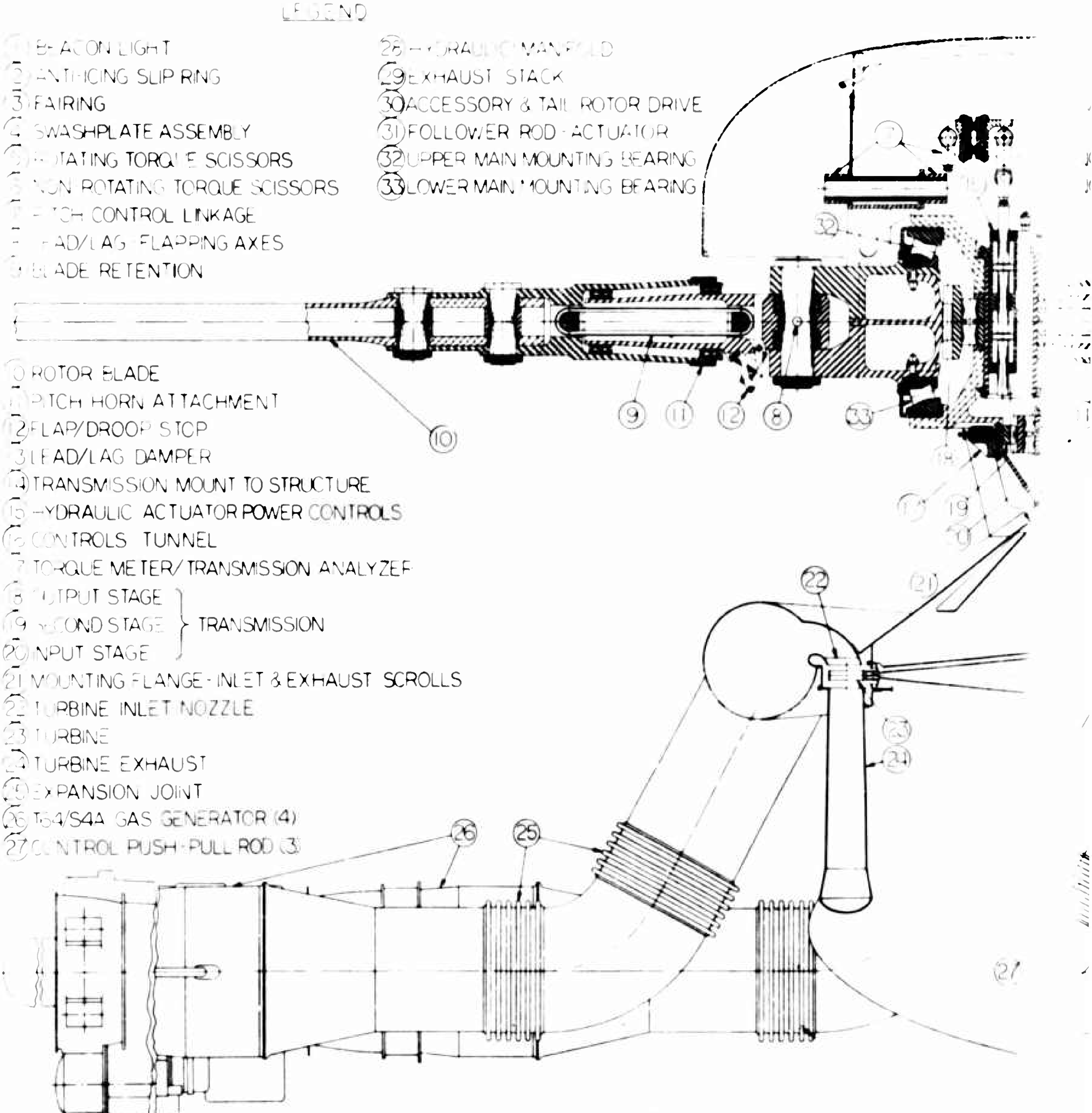
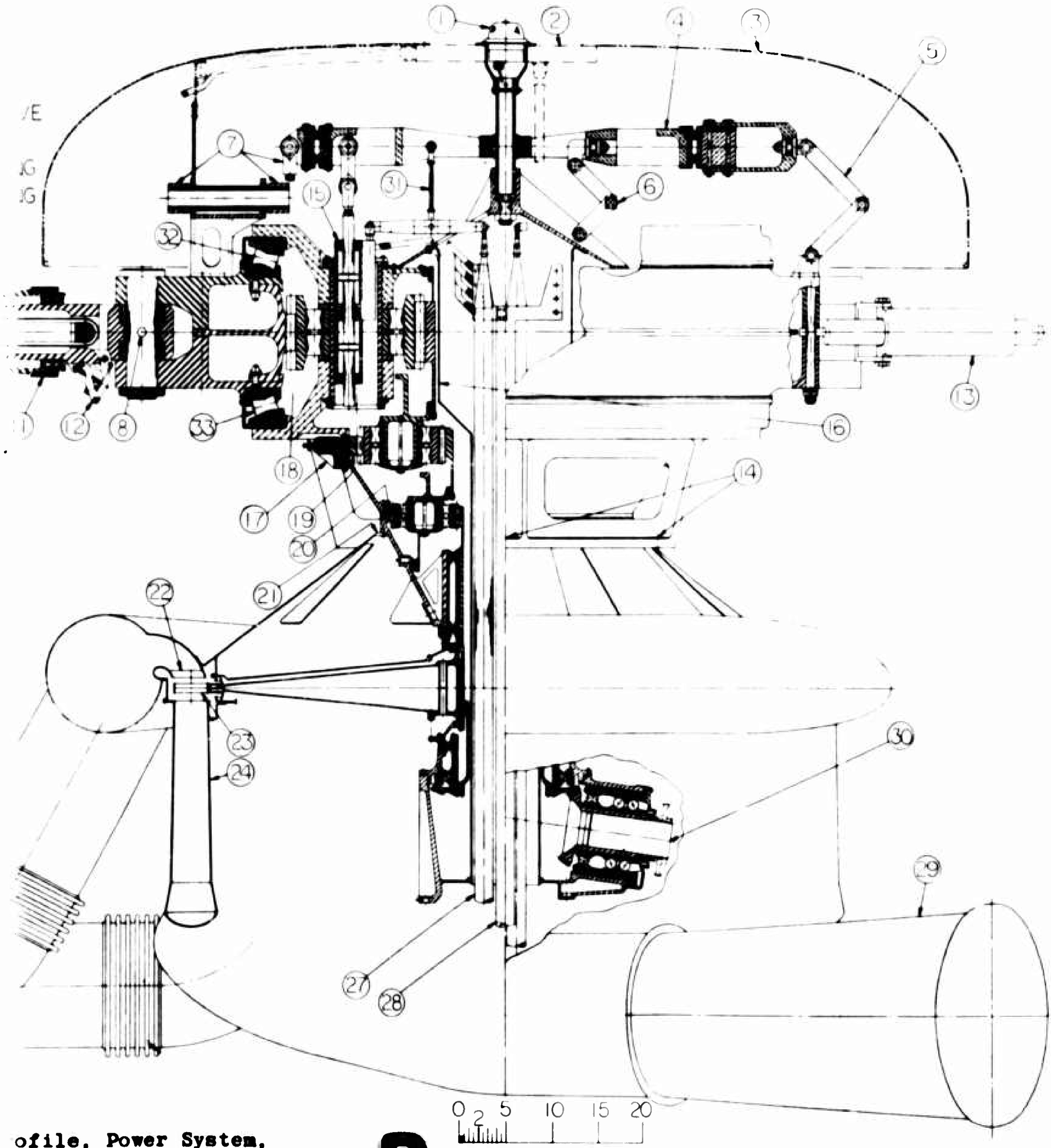


Figure 7. Inboard Profile, Power System, Integrated Rotor Hub



ofile, Power System,
Rotor Hub

B

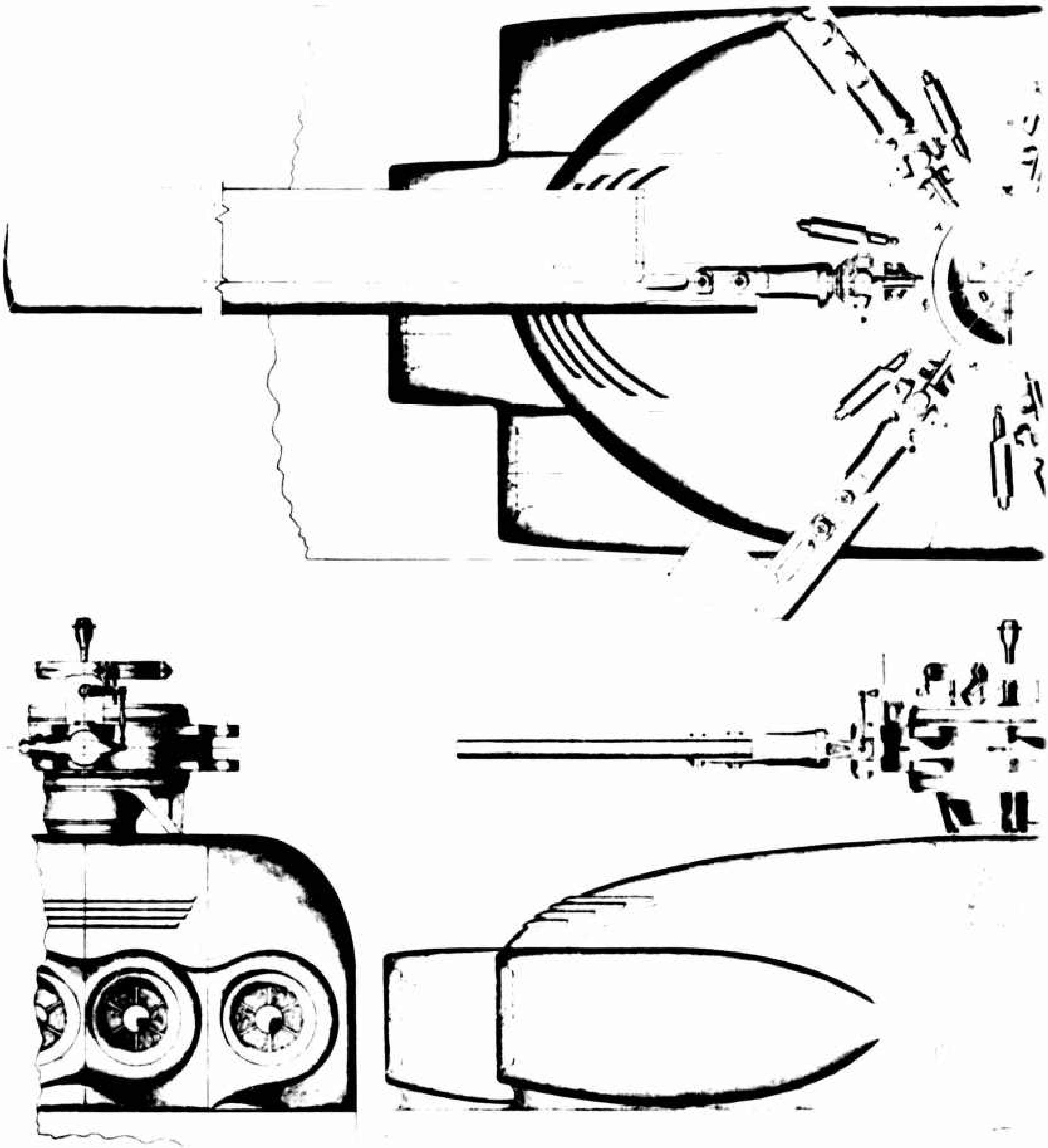
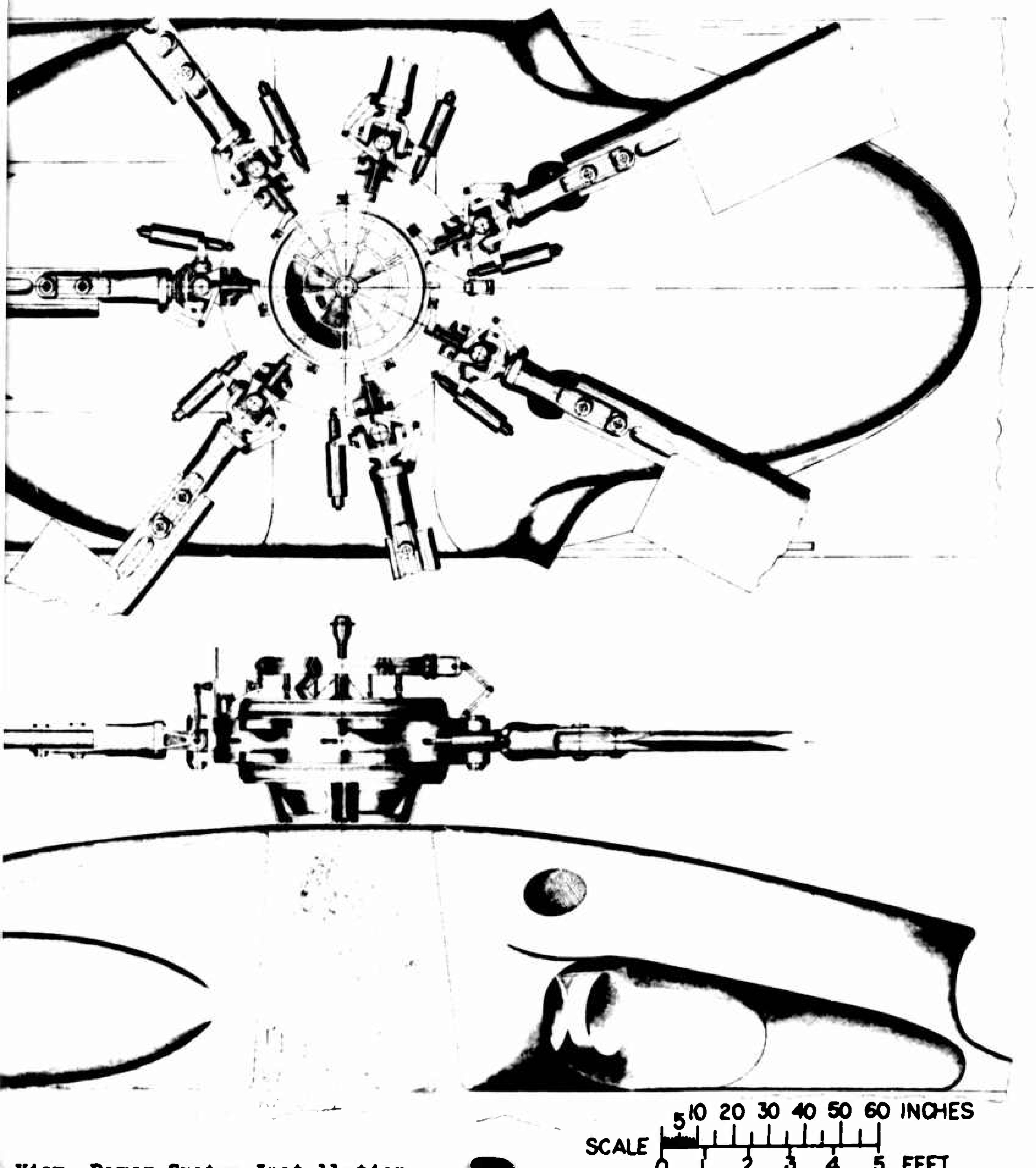


Figure 8. Three-View, Power System Installat
Integrated Rotor Hub

A



-View, Power System Installation,
rated Rotor Hub

B

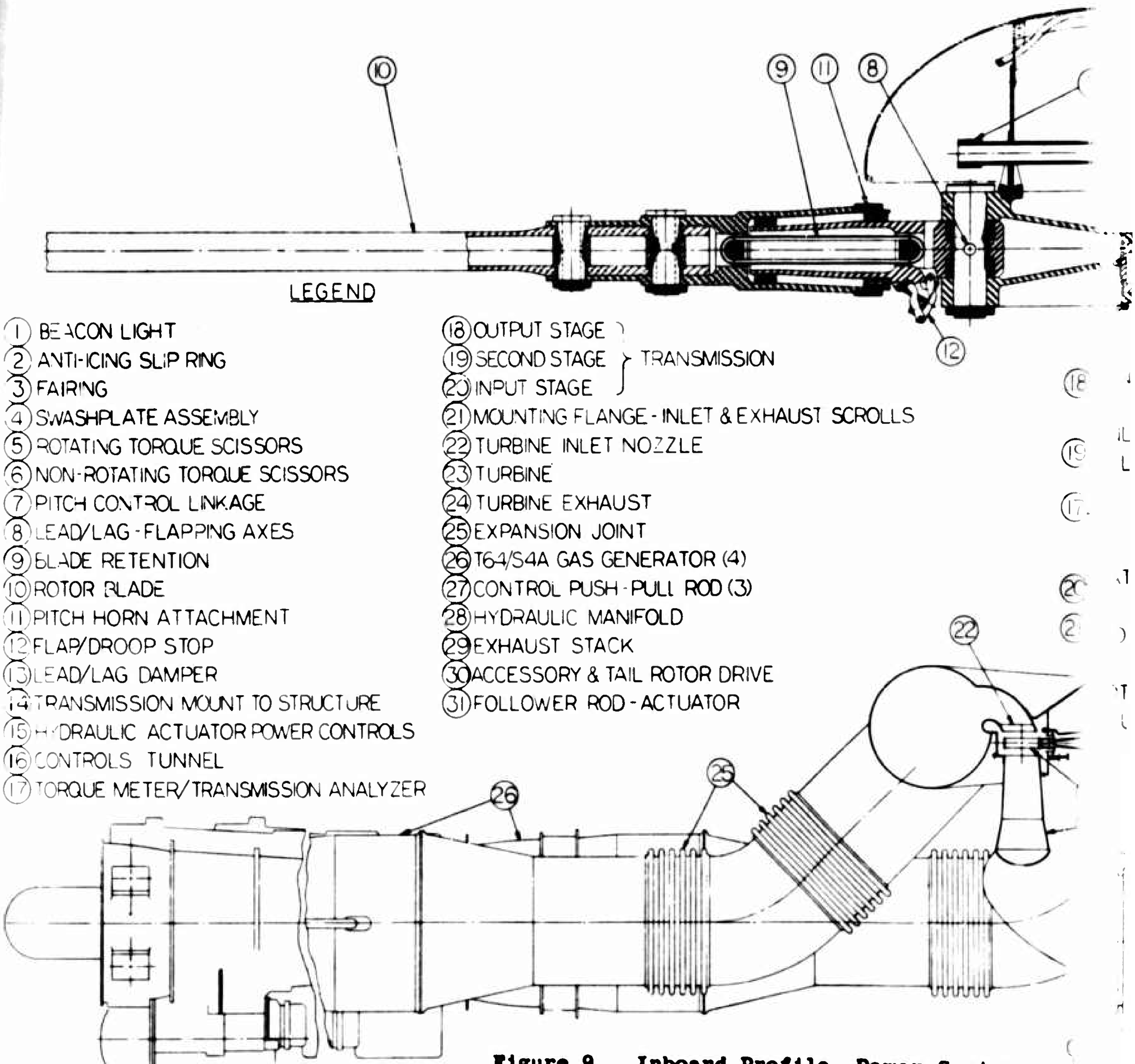
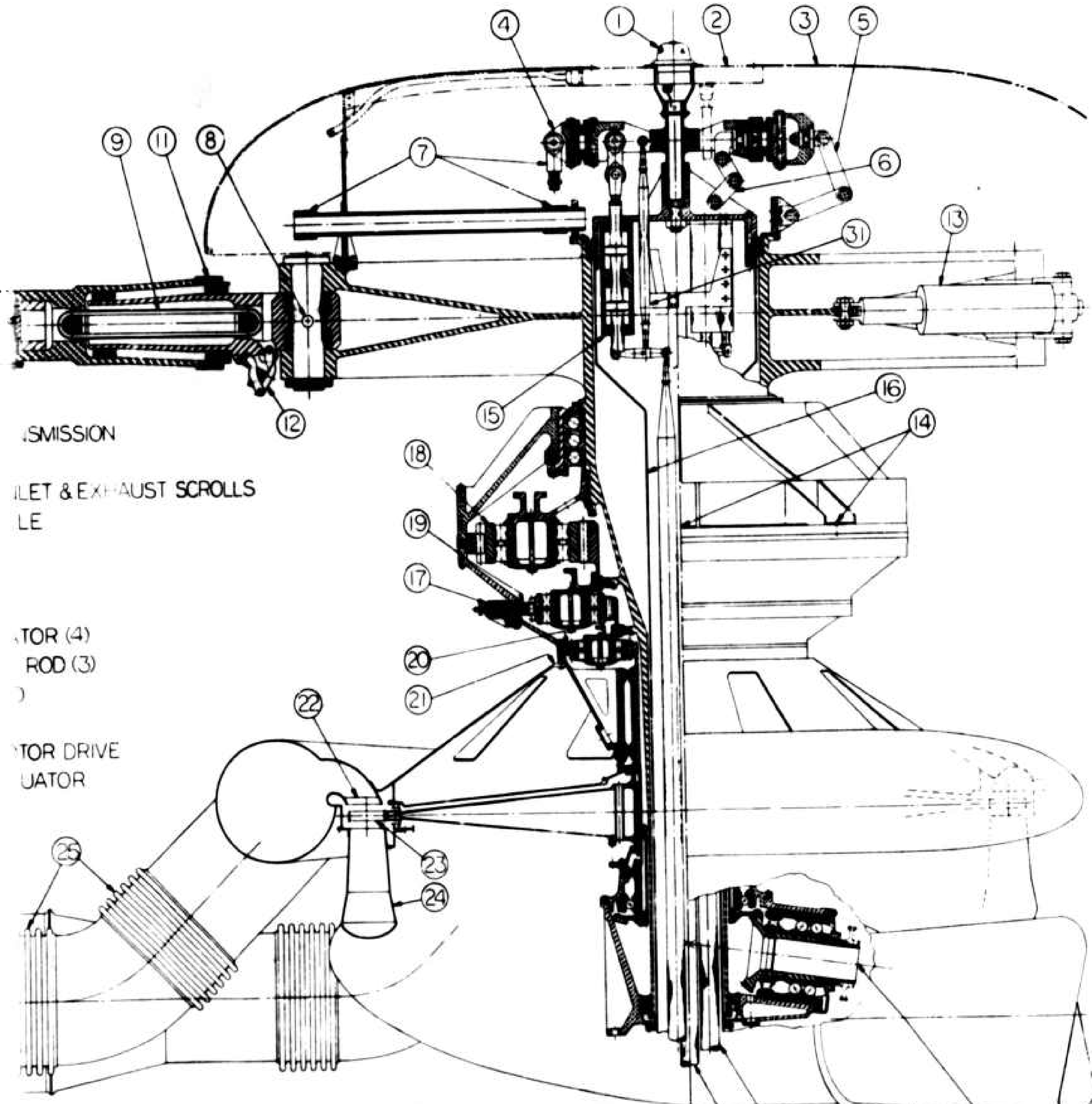


Figure 9. Inboard Profile, Power System, Segregated Rotor Hub



·board Profile, Power System,
-gregated Rotor Hub

B

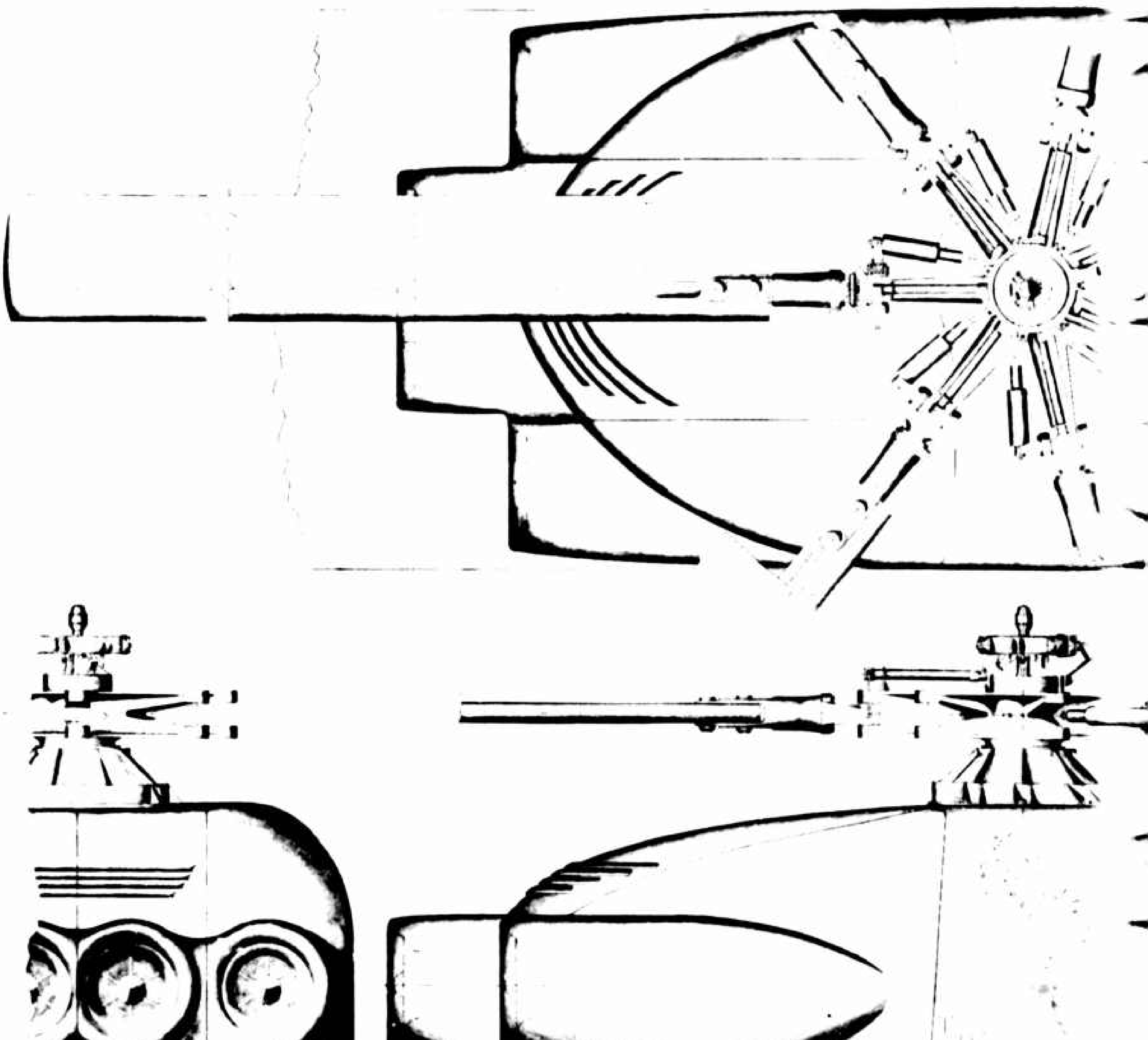
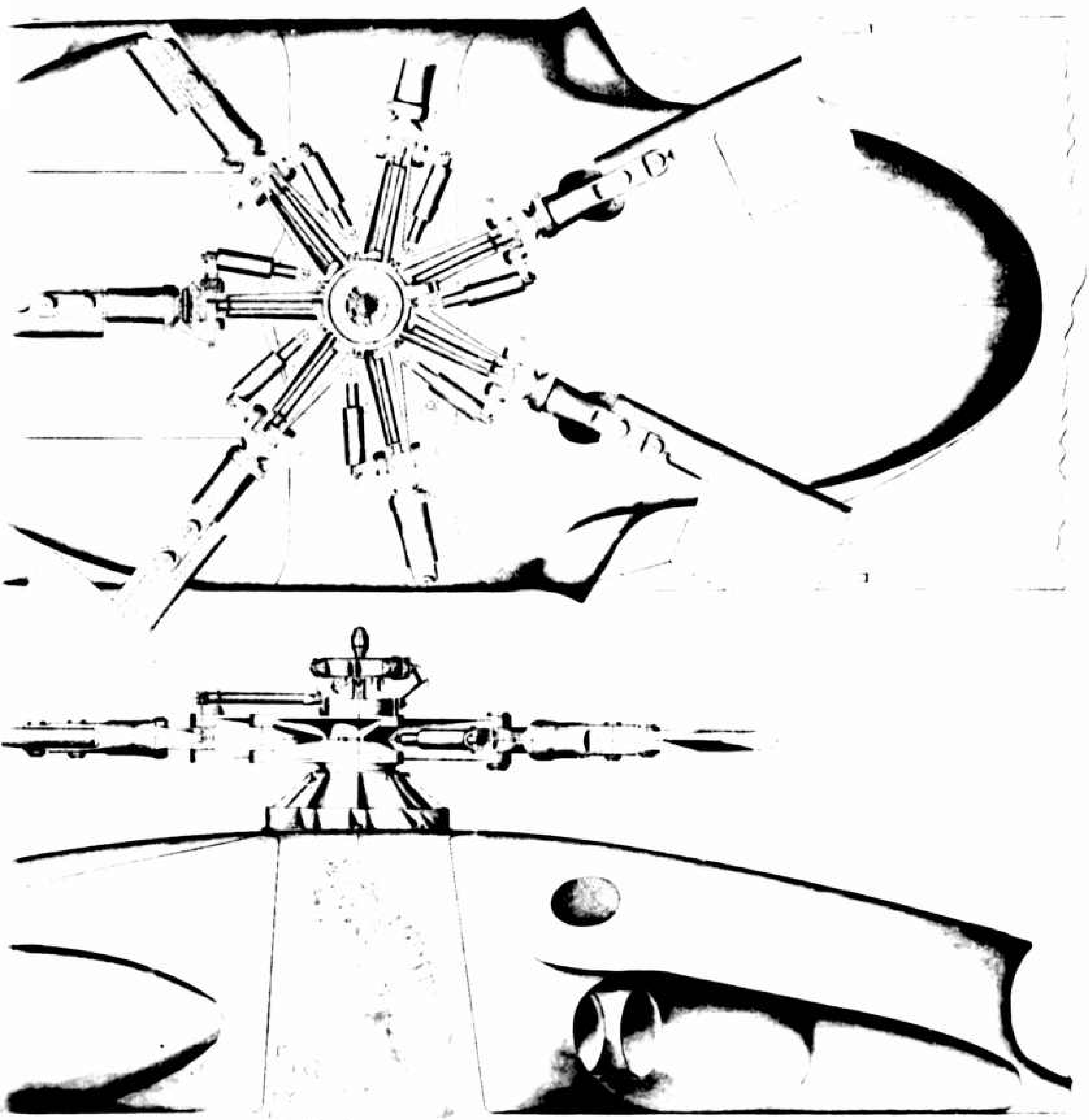


Figure 10. Three-View, Power System Installatio·ee Segregated Rotor Hub ·e

A



ee-View, Power System Installation,
egated Rotor Hub

B

The basic elements and construction techniques of the scrolls and ducting system are similar to those developed for the X-353-5 lift fan system.

The construction and testing of the power turbine and ducting system can be likened to the X-353-5 lift fan without the fan. For programs of comparable magnitude in the development of lift fans, it has been estimated that a demonstration could be accomplished in 14 months from go-ahead, with a PFRT in 18 to 22 months. A development program for the HLH power system, beginning with engine development, is discussed in the section of this report titled Comparative Evaluation of Power Transmission Systems (section C, page 70).

B. ANALYSIS OF POWER TRANSMISSION SYSTEMS

1. Manufacturing Technology

(a) Manufacturing Methods Required - The segregated rotor hub transmission is, in general, simpler to manufacture than is the integrated rotor hub transmission. The integral rotor hub with rotor mast used in the segregated hub system requires the development of a special hob head extension for the precision hobbing machine to cut the spline on the rotor mast. A more serious problem exists with the integrated hub transmission - the internal tooth cutting equipment available at this time cannot accommodate the outside diameter designed for the integral rotor hub/ring gear member. This situation may be relieved within the time scale noted for this program. Some experimental work will be required to determine grades of material and heat treatment, to determine distortion patterns for proper stock allowances for the output stage internal ring gears, and to develop cutting tools for blending tooth root profile with the active tooth profile. The requirement for experimental development is not unusual. It is estimated that four experimental pieces each for the sun and ring gears and eight experimental pieces for the planets will be sufficient to accomplish this development.

(b) Material Availability - Material specifications can be met for all components except the output stage internal ring gears, which can be made of nitrided forgings of material such as vacuum melted 4340 steel.

(c) Gear and Tooth Geometry Limits Including Accuracy Attainable and Deflection Considerations - Specifications

can be met. No problem areas are apparent.

(d) Techniques that Assist Equal Load Distribution Among Multiple Load Paths - Methods are straightforward. No problem areas are apparent.

(e) Stress Distribution and Stress Allowable Data - Stress analysis was performed, using American Gear Manufacturers Association (A.G.M.A.) stress formulae, by a machine program available to the gear manufacturer. With respect to durability horsepower and strength horsepower, the design is conservative. Bending strength in the output mesh is adequate if supported by an experimental material-property investigation for the material substitution recommended. Compressive stress allowables are adequate.

(f) Bearing Geometry and Mounting Requirements - No problem areas are apparent in connection with any of the conventional transmission bearings. The integrated rotor hub main mounting bearing is the subject of special investigation and is reported on page 53 , under title of Deflection.

2. Weight

The two power systems are compared with respect to weight in Table III. Both systems were designed to the same specifications in all respects. Weights were calculated from design layouts after stress analysis. The transmission of the segregated rotor hub system includes rotor mast weight to the top of the upper mast bearing - above this arbitrary boundary the mast weight is included with the hub weight. The hub weight of the integrated rotor hub system does not include the main mounting bearings.

As was anticipated, the integrated rotor hub system with the remote turbine has a lighter total transmission and rotor hub weight than parametric analysis indicates would be found for conventional multiple engine/transmission/rotor hub practice. This study has found, however, that the segregated rotor hub system with the remote turbine is even lighter. The weight saving for the integrated hub is less than the added weight required for the jackshaft plus epicyclic transmission and the mounting bearings. The segregated hub transmission has the benefit of an optimum reduction ratio per stage (3.0+ to 1), and the "framing" technique of rotor hub design.

TABLE III

WEIGHT SUMMARY - MAJOR PARTS AND COMPLETE SYSTEM

Major Parts	Segregated Rotor Hub Power System	Integrated Rotor Hub Power System
	Pounds	
Transmission	3380	6456
Rotor Hub	1931	1731
Transmission and Rotor Hub (Total)	5311	8187
Rotor Blade Retention, Articulation and Control	2707	2707
Affected Portions of Fixed and Rotating Controls	596	584
Affected Portions of Hydraulic System	68	68
Affected Portions of Electrical System	7	7
Affected Portions of Aircraft Structure*	-	-
Main Transmission Oil Cooling System	235	235
Power Plant	3463	3463
Complete Systems (Total)	12387	15251

*Considered as body weight, not as a system components weight

3. Efficiency

(a) Power Plant - Duct losses are estimated to be 3 percent. This is not a power penalty to the aircraft attributable to the remote turbine concept, as an equivalent loss would be sustained by conventional multiple turbine/transmission practice in achieving the same multi-engine combining (4), change of direction (90 degrees), and speed reduction (13600 r.p.m. to 3600 r.p.m.). Power and SFC figures for this engine are based on power delivered by the remote turbine at the transmission interface and include the duct loss.

Losses that may be attributed legitimately to the remote turbine concept include disc windage (constant), shroud windage (increases as gas generators are removed), seal leakage (decreases as gas generators are removed), inactive/active quadrant interface losses (increases, then decreases as gas generators are removed), and idle bucket windage (increases as gas generators are removed). It will be observed from Figure 15 that a lower specific fuel consumption is achieved for a reduced power requirement by removing a gas generator than results from four gas generators operating at the same reduced power level.

(b) Transmission - Gear efficiency (see Table IV) was calculated according to the method presented by Dudley, "Practical Gear Design", reference 4, section 14, pages 24-26. Bearing efficiency was calculated according to the method presented by the same reference, section 14, pages 17-19. Efficiency calculations are notoriously imprecise, and are presented only as a method of comparative evaluati n of two transmissions.

TABLE IV

COMPARISON OF REDUCTION GEARING EFFICIENCY

Parts	Segregated Rotor Hub Transmission Percent	Integrated Rotor Hub Transmission Percent
Input Stage Gear Efficiency	99.410	99.280
Second Stage Gear Efficiency	99.395	99.300
Output Stage Gear Efficiency	99.395	99.250
Total Gear Efficiency	98.210	97.840
Input Stage Bearing Efficiency	99.000	98.960
Second Stage Bearing Efficiency	99.130	99.030
Third Stage Bearing Efficiency	99.080	98.940
Total Bearing Efficiency	97.240	96.960
Total System Efficiency	95.500	94.360

4. Stress

Gear tooth stresses are for 12000 horsepower at design speeds for the main transmission and 1920 horsepower at 3600 r.p.m. input for the spiral bevel gear set. These stresses are calculated by the following methods:

(a) Bending - "High Capacity Gearing", reference 2, pages 130-160.

(b) Hertz - The conventional Hertz equation.

(c) Scoring - Kelley equation, "Practical Gear Design", reference 5, pages 141-142.

Bearing life calculations are calculated by American Standards Association (A.S.A.) method, "Load Ratings for Ball and Roller Bearings", reference 1, pages 6-12, except for the integrated rotor hub mounting bearings, which were calculated under subcontract by a machine program. The following load schedule was used for bearing life calculations:

(a) Planetary Bearings - 12000 horsepower at design speed.

(b) Rotor Mounting Bearings - Continuous maximum combined rotor thrust and hub moment.

(c) Turbine Bearings - Worst continuous case of thrust and moment, MIL-E-8593 (ASG) accelerations.

(d) Spiral Bevel Bearings - 0 hours at 2400 horsepower, 180 hours at 1920 horsepower, 840 hours at 1080 horsepower, 180 hours at 480 horsepower, all at 3600-r.p.m. input, 7200 r.p.m. output.

The mast rotating bending stress was calculated by conventional methods. Hub stresses were calculated by a variation on the methods presented by Kleinlogel, "Rigid Frame Formulas", reference 7, pages 151-153.

TABLE V
STRESS SUMMARY

IRHT*	Calculated Bending (p.s.i.)	Calculated Hertz (p.s.i.)	Calculated Scoring °F (Blank Temp. 180°F)	Calculated B-10 Life, Load Conditions Noted (Hours)
Input Stage				
Sun	40,000	132,000	376	-
Planet/Bearing	-5500+24,700	132,000	376	1,300
Ring	13,000	120,000	328	-
Second Stage				
Sun	33,000	145,000	415	-
Planet/Bearing	-5375+25,000	145,000	415	1,450
Ring	12,600	130,000	330	-
Output Stage				
Sun	24,000	145,000	**	-
Planet/Bearing	-4500+25,000	145,000	**	1,690
Ring	13,900	129,000	**	-
Spiral Bevel Set				
Pinion	30,200	147,000	-	-
Gear	30,200	147,000	-	-
Main Rotor Mtg. Brg.				
Upper	-	-	-	15,000+
Lower	-	-	-	15,000+

* Integrated Rotor Hub Transmission

** Flash temperatures for the output stage, as compared to the second stage with relatively the same Hertz stresses, are not critical by reason of the slower speed at the final stage.

TABLE V (Continued)

IRTH*	Calculated Bending (p.s.i.)	Calculated Hertz (p.s.i.)	Calculated Scoring °F (Blank Temp. 180°F)	Calculated B-10 Life, Load Conditions Noted (Hours)
Turbine Mounting Bearings				
Upper	-	-	-	7,180
Lower	-	-	-	7,180
Spiral Bevel Pinion Bearings				
Thrust	-	-	-	2,600
Radial	-	-	-	5,250
Spiral Bevel Gear Bearings				
Thrust	-	-	-	9,450
Radial	-	-	-	2,570
Hub - Worst Case	$56,500 \pm 11,310$	-	-	-
Allowable	$\pm 25,000$ $+40,000$	150,000	425°F	1,200 Hours

* Integrated Rotor Hub Transmission

TABLE V (Continued)

SRHT*	Calculated Bending (p.s.i.)	Calculated Hertz (p.s.i.)	Calculated Scoring °F (Blank Temp. 180°F)	Calculated B-10 Life, Load Conditions Noted (Hours)
Input Stage				
Sun	39,600	145,000	413	-
Planet/Bearing	-5,550+24,800	145,000	413	1,340
Ring	12,500	129,000	416	-
Second Stage				
Sun	31,800	145,000	346	-
Planet/Bearing	-5,350+24,950	145,000	346	1,865
Ring	13,000	142,000	367	-
Output Stage				
Sun	32,600	144,800	**	-
Planet/Bearing	-4,520+25,000	144,800	**	1,550
Ring	13,500	141,000	**	-
Spiral Bevel Set				
Pinion	30,200	147,000	-	-
Gear	30,200	147,000	-	-
Main Rotor Mtg. Brg.				
Thrust	-	-	-	1,490
Radial	-	-	-	3,300

* Segregated Rotor Hub Transmission

** Flash temperatures for the output stage, as compared to the second stage with relatively the same Hertz stresses, are not critical by reasons of the slower speed at the final stage.

TABLE V (Continued)

SRHT*	Calculated Bending (p.s.i.)	Calculated Hertz (p.s.i.)	Calculated Scoring °F (Blank Temp. 180°F)	Calculated B-10 Life, Load Conditions Noted (Hours)
Turbine Mounting Brgs.				
Upper	-	-	-	7,180
Lower	-	-	-	7,180
Spiral Bevel Pinion Bearings				
Thrust	-	-	-	2,600
Radial	-	-	-	5,250
Spiral Bevel Gear Bearings				
Thrust	-	-	-	9,450
Radial	-	-	-	2,570
Mast - Worst Case	$\pm 24,700$	-	-	-
Hub - Worst Case	$56,500 \pm 11,310$	-	-	-
Allowable	$\pm 25,000$ +40,000	150,000	425°F	1,200 Hours

* Segregated Rotor Hub Transmission

5. Deflection

The deflection consideration of importance in this study was concerned with the mounting bearings for the integrated rotor hub/ring gear. The mounting bearings must withstand a seven-lobed tri-axial deflection from rotor blade loads superimposed on a six-lobed bi-axial deflection from gear loads. The two multi-lobed patterns rotate with respect to each other. The deflections were determined by energy methods for five individual sources of load. The deflection from two loads were by hand calculation and the deflection from three loads were by a machine program developed for the structural analysis of fuselages. The superimposed deflections are shown on Figures 11, 12, 13 and 14 for both bearings for two instances in time, at t = 0 and at t = 0 + 1/14 rotor revolution, as examples of the results of the analysis.

6. Dynamic Behavior

(a) Inertia Loading - The turbine is the item of interest. Load conditions calculated include force from maneuver acceleration, moment from gyroscopic precession, and force and moment resulting from inactive gas input quadrants.

(1) Maneuver Accelerations*

Force (F) = mass (m) x acceleration (a)

Angular acceleration = 12 rad/sec^2 about aircraft center of gravity, a point approximately 7.5 ft below the turbine centerline plane.

$a = 12\ \text{rad/sec}^2 \times 7.5\ \text{ft} = 90\ \text{ft/sec}^2$

m = 252 lb turbine + misc shaft weight, use 300 lb/32.2 ft/sec^2 = 10 lb sec^2/ft

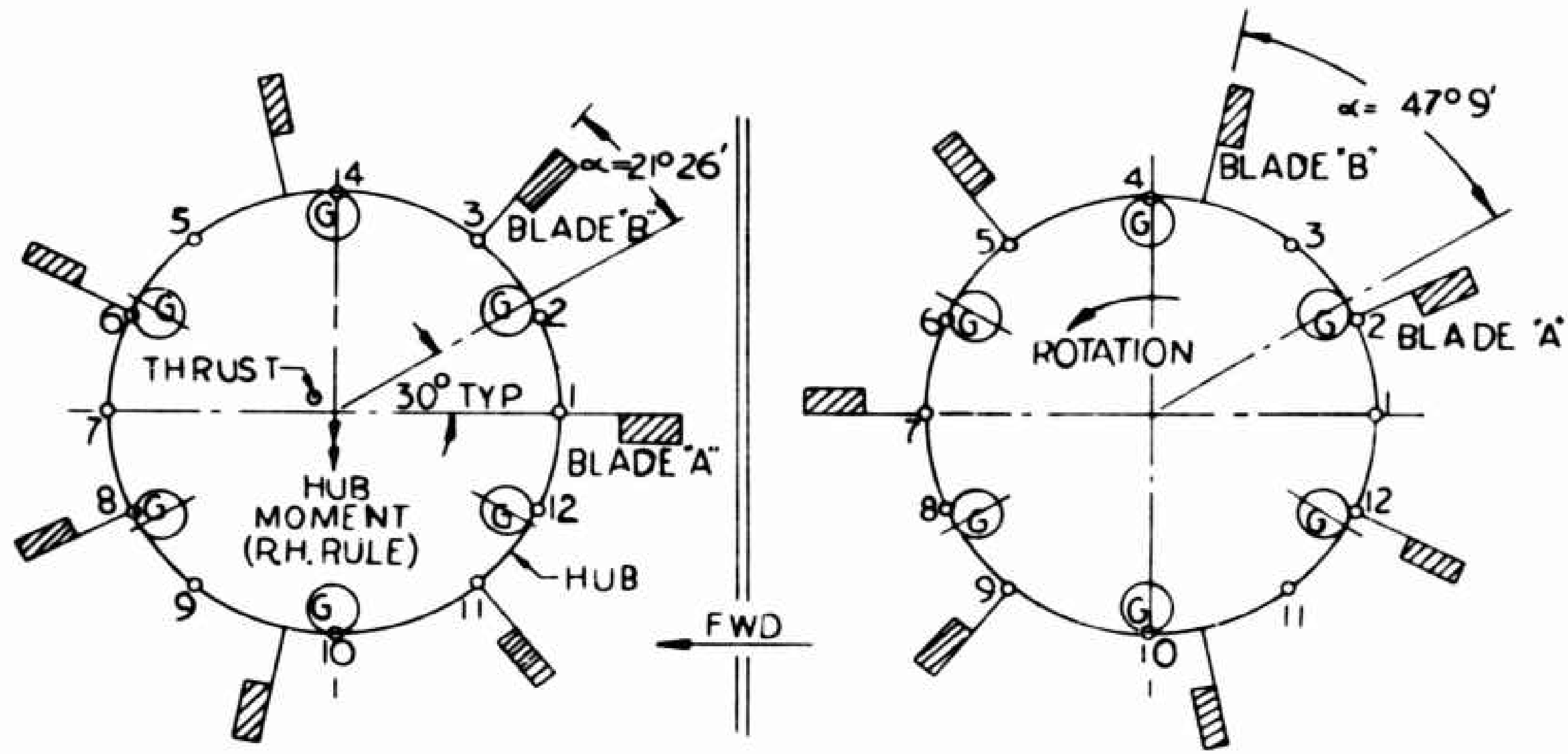
$F = ma = 90 \times 10 = 900$ lb at turbine centerline plane

(2) Gyroscopic Precessional Moment*

Moment (M) = polar mass moment of inertia x angular acceleration (12 rad/sec^2 per MIL-E-8593 (ASG))

$M = (443\ \text{lb-in sec}^2)\ (12\ \text{rad/sec}^2) = 5300\ \text{lb-in}$

*Both force and moment present transient load cases to the turbine mounting bearings as well as transient rotating bending stress to the turbine and turbine shaft.



TIME = 0

TIME = 0 + 1/4 OF A ROTOR REV.

Figure 11. Plan View of Integrated Rotor Hub Main Mounting Bearings

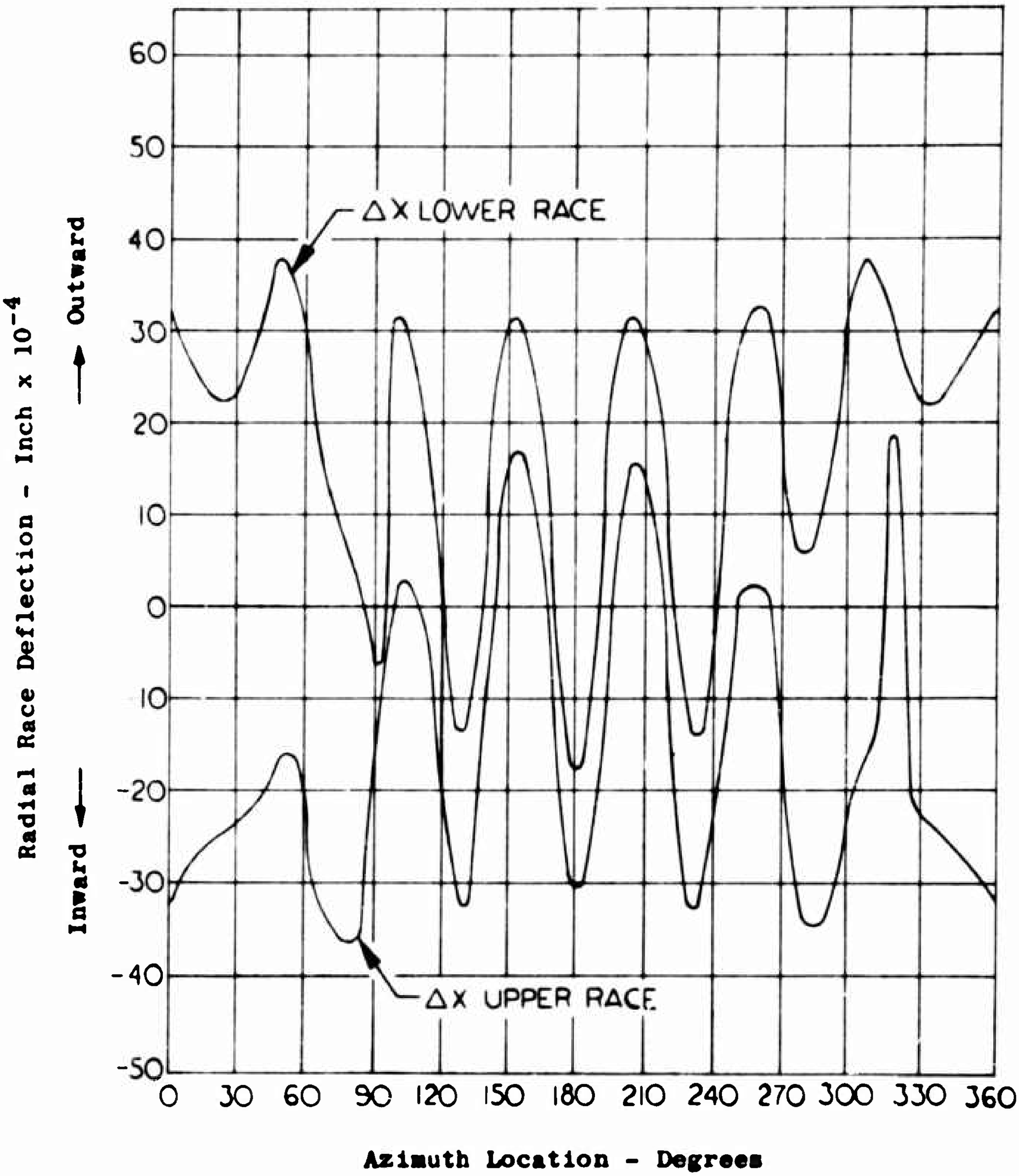


Figure 12. Radial Bearing Deflection Distribution, Time - 0

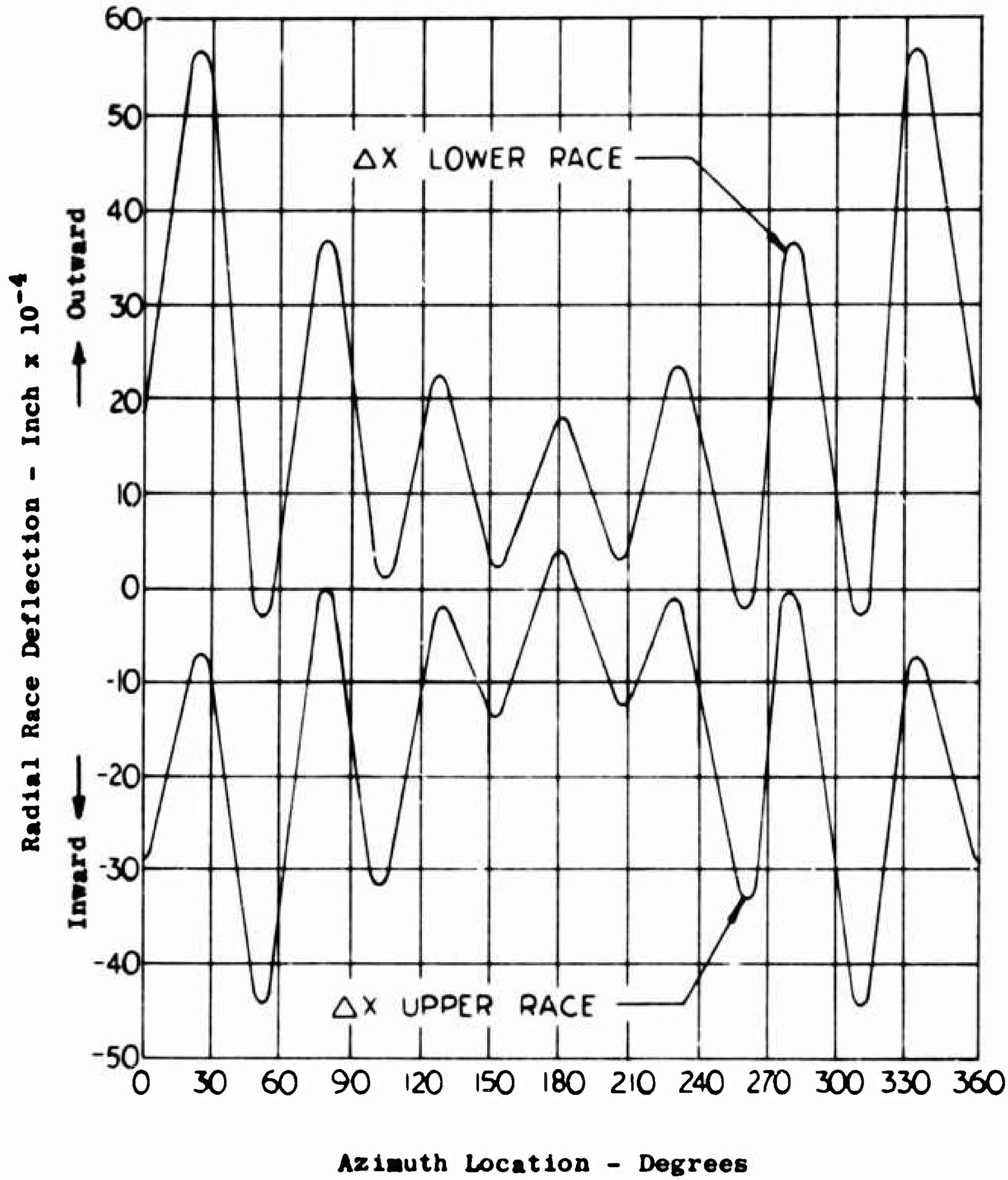
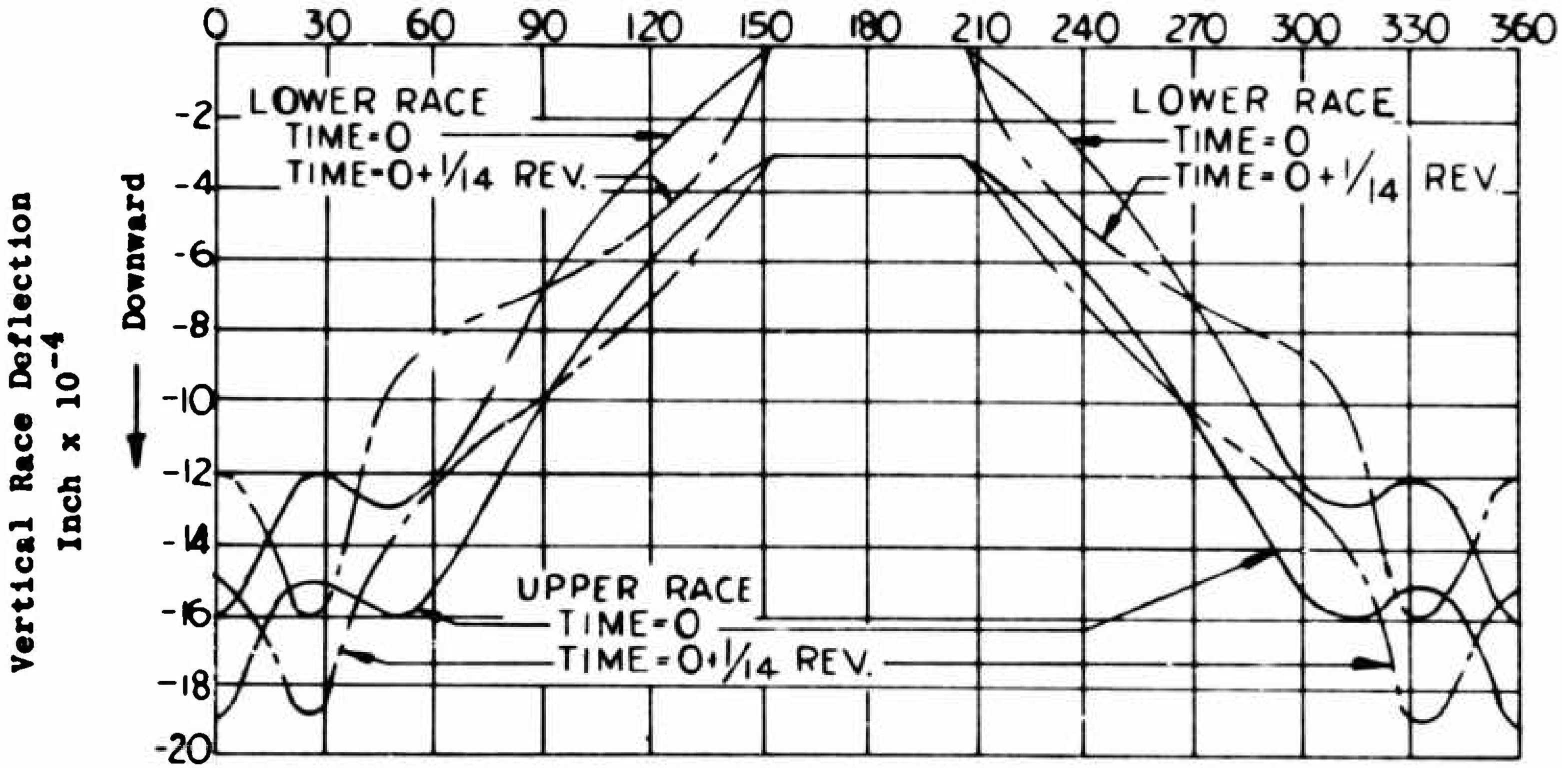


Figure 13. Radial Bearing Deflection Distribution, Time = 0 + 1/14 Rotor Revolution



Azimuth Location - Degrees

Figure 14. Vertical Bearing Deflection Distribution

(3) Force and Moment Resulting From Inactive Gas Input Quadrants

Equations were derived to find the force centroid ($\bar{r}$) when partial gas entry occurs over an included angle (θ) for all possible cases of inactive gas generators. For all cases, the equation is:

$\bar{r} = 48.7\ (1-\cos\ \theta)^{\frac{1}{2}}/\cos\ \theta$.

Axial forces and moments are presented in the following table (Table VI). Two adjacent gas generators operating is the case that was used as the steady state operating condition for turbine mounting-bearing life calculation.

TABLE VI

TURBINE FORCE AND MOMENT SUMMARY

Case	Axial Force Pounds	Moment Pound-Inches
4 Gas Generators Operating	+800	-
3 Gas Generators Operating	+600	6200
2 Adjacent Gas Generators Operating	+400	8770
2 Opposite Gas Generators Operating	+400	-
1 Gas Generator Operating	+200	6200
0 Gas Generators Operating	Negligible	-

(b) Natural Frequency of Rotating Parts - There are two areas of concern: turbine behavior and hypercritical shafting for the tail rotor drive. A discussion on turbine natural frequencies and dynamic behavior is given on page 33 of this report, under Turbomachinery Deflection and Dynamics. The transmission driven by the turbine features a hunting planet mesh to avoid tooth impact frequencies at a level of interest to the turbine. In addition, each planetary stage is indexed to be out of resonance with the other stages. The hypercritical shafting for the 7200-r.p.m.

case selection operates to the 16th critical speed (7940 r.p.m.) and is stable to the 17th critical speed (8800 r.p.m.). Shaft design speed is 7080 r.p.m.

(c) Noise - Fundamental Frequencies - The purpose of this identification is to support any future investigation of the noise environment of the HLH. It is believed that gearbox frequencies are widely scattered, providing "white" noise. Components and operating speeds are identified in Table VII.

TABLE VII

FUNDAMENTAL FREQUENCIES OF ROTATING PARTS

Component	Operating Speed
Gas Generators	13600 r.p.m.
Tail Rotor Drive Shaft	7080 r.p.m.
Turbine	3600 r.p.m.
Rotor *	124 r.p.m.
Rotor Tip Speed	650 ft/sec
Tail Rotor *	655 r.p.m. (Dual Rotors) 478 r.p.m. (Single Rotor)
Tail Rotor Tip Speed	650 ft/sec

*Numbers of blades on the rotor and tail rotor are not definite, pending the outcome of the rotor study.

7. Reliability

Reliability is shown on Table VIII for individual parts, assembled from data on parts in service, and systems, calculated by the process shown. The parts specified for conventional multiple turbine/transmission practice are believed to be at a minimum. Schematically, four turbines each would drive with 3090 horsepower at 13600 r.p.m. through a misalignment coupling device to a freewheeling clutch connected to an input pinion. The four input pinions would engage one combining bull gear. The bull

gear would power a drive shaft at 6000 r.p.m., with the drive shaft connected to a change-of-direction gear set. The output of the change-of-direction gear set is assumed to be 12000 horsepower at 3600 r.p.m., equal to the output of the remote turbine. The remaining speed reduction is assumed to be identical in both cases, corresponding to the gearbox investigated for the remote turbine/segregated rotor hub transmission. The result is

percent improvement in MTBF when failure means mission abort =

(Conv/TIGR-1) 100 = (.001416/.000376-1) 100 = 280 percent.

The failure rates (mission abort) used in this study were determined as follows:

(a) The engine manufacturer supplied the failure rates of the items for which they were responsible or for which they had values available.

(b) Items that are similar to the UH-2 used UH-2 failure rates; these rates are based on failures that occurred during 30000-flight hours. If no failures were reported, the failure rates were calculated based on total operating hours (30000) of the system or component and a confidence level of 90 percent.

(c) The failure rates for the remaining items were obtained from the U. S. Naval Ordnance Laboratory.

TABLE VIII

COMPARISON OF RELIABILITY

Components	No.	CONV. TURBINES WITH GEARBOX (Parts Failure Rate/Flight Hour)	No.	REMOTE TURBINE WITH GEARBOX (Parts Failure Rate/Flight Hour)
Gas Generators	4	.000006*	4	.000006*
Turbines With Brgs.	4	.00004	-	-
Cruise Fan Turbine With Bearings	-	-	1	.00001
13600-R.P.M. Seals	4	.00004	-	-
Misalignment Couplings	4	.00032	-	-
Freewheeling Unit Sprag Clutch With Bearings	4	.00032	-	-
Freewheeling Units - Variable Stators	-	-	4	**
Engine Combining Mesh With Bearings	1	.00020	-	-
Drive Shaft	1	.00008	-	-
Change of Direction Mesh With Bearings	1	.00001	-	-
Epicyclic Speed Reduction Meshes	3	.00024	3	.00024
Oil System Size and Complexity	2	.00008	1	.00004
124-R.P.M. Seal	1	.00001	1	.00001
Tail Rotor and Accessory Drive w/Brgs.	1	.00007	1	.00007
Totals	30	.001416	15	.000376

*This is the equivalent failure rate for the ferry mission (13.7 hr). 2 of the 4 gas generators are required to operate.
**Included with cruise fan failure rate.

8. Maintainability

The TIGR HLH configuration offers several maintenance advantages over a system combining conventional turbine engines with a typical mechanical drive arrangement. These advantages result from fewer primary drive components than in a conventional design.

The TIGR concept simplifies the engines and eliminates the drive shafting and intermediate gearing required for speed reduction and change of direction. This necessitates fewer component replacements and eliminates the need for time-consuming shaft alignment with engine and gearbox changes. Benefits also accrue in logistics support requirements, since fewer high-value critical components need be stocked and cycled through overhaul.

Two relatively new devices will add measurably to the safety and reliability of the TIGR HLH and provide attendant improvements in maintainability. These devices are the rotor overspeed recorder and the torquemeter/transmission analyzer. Prescribed maintenance practice following occurrence of a known or suspected rotor overspeed involves complete inspection or removal and disposal of rotor components. This is an expensive procedure, both in terms of downtime and support costs, particularly when such action is taken to avoid possible risk when the extent of overspeed is not actually known. The rotor overspeed recorder will eliminate this eventuality.

The second device, the torquemeter/transmission analyzer, offers a unique method of measuring transmission operating life. By monitoring operating intervals in terms of load exposure rather than operating hours, it promises to extend the period between replacement with consequent savings in man-hours, downtime, and logistic support costs.

(a) Maintenance Analysis - A preliminary analysis of the TIGR and power train layout proposed for the heavy lift helicopter has been made to estimate the maintenance man-hour per flight-hour requirement at first through third maintenance echelons. Included in this analysis were the gas generators, remote turbine/transmission assembly, accessory gearbox, auxiliary power unit, and accessory components. The study involved a detailed task analysis of routine maintenance requirements, including servicing, inspections, and scheduled component replacement. The results of this study, shown below, indicate that the systems evaluated can be supported at the organizational

and direct maintenance levels with approximately two and one-half maintenance man-hours per flight-hour.

TABLE IX

ESTIMATED MAINTENANCE MAN-HOURS PER FLIGHT HOUR
(TIGR HLH ENGINE AND PRIMARY DRIVE SYSTEMS)

Echelons	1st	2nd	3rd	Total
Servicing	.099	-	-	.099
Daily Inspection	-	.576	-	.576
Intermediate Inspection	-	.091	-	.091
Periodic Inspection	-	-	.080	.080
Scheduled Component Replacement	-	-	.168	.168
Unscheduled Maintenance	-	.608	.913	1.521
Totals	.099	1.275	1.161	2.535

Factors Used For Analysis:

(a) 2 1/2-hours-per-day average utilization
(b) 25-hour intermediate inspection interval
(c) 100-hour periodic inspection interval
(d) 600-hour gas generator and APU TBO
(e) 1200-hour gearbox TBO
(f) historical data for unscheduled maintenance

(b) Design Considerations - Design progress has been monitored for maintainability considerations with particular emphasis on accessibility, arrangement, and location of components. The nacelle cowling is constructed in two large sections which move forward and aft on tracks to expose the entire engine-power train package. Large work areas on the cabin roof provide ample access to all components. Access to the rotor head is provided by large work platforms mounted to the frame structure which supports the turbine/transmission assembly. The platforms are reached by three permanent steps on each side of the structure. For a standing mechanic, the rotor hub components are at workbench height from these platforms, as shown in the power plant installation drawing (Figure 6).

A major cause of premature replacement of helicopter transmissions is seal leakage. It is planned to use a split seal and split seal retainer for the upper mast seal on the segregated rotor hub transmission to allow seal replacement without having to remove the transmission. The lower shaft seal on both transmission designs is by static O-ring, with no wear life limit.

It is anticipated that the majority of maintenance tasks will be accomplished with standard tools and equipment available to third echelon maintenance units. The only requirements for special tools foreseen at this time are wrench adapters to be used with the Army pin spanner and hydraulic wrench for high-torque bolt applications and gearbox/gas generator lifting slings. Incorporation of an auxiliary power unit (APU) eliminates requirement for additional ground support equipment (GSE) while enabling complete checkout and trouble-shooting of hydraulic and electrical/electronic systems without the use of the main drive system.

(c) Maintainability Program Plan - The TIGR configuration will be continually reviewed and analyzed for maintainability considerations during any future design/development program. Guidelines will be provided to the designers in the areas of accessibility, removal/installation, repair, assembly/disassembly and support equipment minimization and compatibility. Analytical studies will be performed to statistically predict such factors as mean/maximum downtime, maintenance man-hours per flight-hour and overhaul turn-around time. During the course of the qualification test program, maintenance tasks will be demonstrated to the extent desired to substantiate the predictions.

9. Costs

HLH system costs are summarized in Table X, below. The values shown indicate qualitative relationships rather than precise quantitative data. These cost figures are based on assumptions discussed in later paragraphs. Total cost is taken as the sum of procurement cost plus maintenance cost plus petroleum, oil, and lubricant cost plus crew cost plus attrition cost in dollars per flying hour.

TABLE X

HEAVY LIFT HELICOPTER COSTS PER FLIGHT HOUR*

	Standard Transmission Dollars/Hour	Integrated Rotor Hub Transmission Dollars/Hour	Segregated Rotor Hub Transmission Dollars/Hour
Procurement	$2390 (52.4%)	$2400 (51.8%)	$2280 (53.1%)
Maintenance	$ 310 (6.8%)	$ 315 (6.8%)	$ 295 (6.9%)
Petroleum, Oil, Lubricants	$ 890 (19.5%)	$ 950 (20.5%)	$ 850 (19.8%)
Crew	$ 20 <(1.0%)	$ 20 <(1.0%)	$ 20 <(1.0%)
Attrition	$ 950 (20.8%)	$ 950 (20.5%)	$ 850 (19.8%)
Total Cost	$4560 (100%)	$4635 (100%)	$4295 (100%)
*Procurement Cost/Helicopter	**$8,600,000**	**$8,660,000**	**$8,220,000**

(a) Assumptions Relating to Costs - Unless otherwise specified, assigned values of cost have been predicated upon the basic data of reference 3, pages 36-39 and 42-48.

(b) Procurement - Research, development, test, evaluation and production are included herein. The production quantities discussed in reference 3 refer to 500 units and are therefore not germane to the HLH. An industry yardstick of $100 per pound of gross weight has been utilized. To find procurement cost per flight hour, the gross weight multiplied by $100 per pound is divided by the 3600 flying hours of service life specified for this study. No residual value is assumed.

(c) Maintenance - The original data were based on reciprocating engine-powered helicopters. Lower values resulting from use of turbine engines has been assumed to be balanced by the added navigational, control augmentation, and other subsystems found in modern aircraft. Data were therefore used without change.

(d) Petroleum, Oil and Lubricants - A distorted picture is obtained if we ignore the cost of fuel, as delivered to an overseas base, that is, logistic cost. Public figures for this cost (relating to Vietnam) have ranged as high as $15.00 per gallon. In the interests of conservatism, we can use $1.50 per gallon. Since this is ten times the cost of fuel (15 cents per gallon) at the source, data from the appropriate curves have all been multiplied by a factor of ten. Thus, the sensitivity of petroleum, oil and lubricants (and total operating) costs to SFC will be indicated.

(e) Crew - This is a fixed value based upon crew size, rather than type of transmission or helicopter. It is negligible and can be ignored.

(f) Attrition - This is defined as vehicles lost because of weather, component malfunction, or pilot error. Losses due to enemy action are not included. Reference 3 shows an exponential increase in attrition cost per flight hour with increasing gross weight, reflecting increased aircraft complexity and mission difficulty as size increases. The use of these data would unfairly penalize the standard configuration with the high gross weight, as the HLH comparison here is for aircraft of similar complexity and performing the same missions. Therefore, attrition cost per flight hour is distributed by weight ratio, using reference 3 data for the attrition cost per flight hour for the lightest aircraft only. Implicit in this method is the assumption that attrition rates for all three aircraft are identical, conservatively neglecting the increased reliability of the TIGR power system, because total HLH reliability figures are not available.

10. Development Time

Development time estimation requires consideration of a regenerative test rig, a power absorption rig, and a compatibility test program.

(a) The Regenerative Rig - The regenerative rig makes it possible to arrive at possible problem areas earlier in the PFRT, MQT, and endurance test programs, because of the duplication of tested components. The object of the test is to demonstrate sufficient integrity to establish the specified 3600-hours service life and 1200-hours time between overhaul (TBO). An indefinite life will be affirmed by analytical overhaul.

Two test transmissions will be run back-to-back in a locked-in torque regenerative test rig. The input, output, and tail rotor-accessory drive shafts of one transmission will be locked to their corresponding shafts on the other. Torque, hub moment, shear, and thrust will be applied in controllable magnitudes in such a manner as to duplicate the loads in both transmissions. The rig is capable of driving the transmission at speeds and torques as specified in MIL-T-8679.

The test is required to demonstrate infinite life for all components except bearings which must demonstrate adequate life for initial TBO conditions. Infinite life is demonstrated by the accumulation of 30x 10^6 cycles on all components except bearings at normal rated power operating conditions (see reference 11, pages 13-19). Comparing the MIL-T-8679 test schedule, reference 10, paragraph 3.6.3.3 with continuous power operation at normal rated torque results in one test hour at normal rated power being equivalent to 2.0-MIL-T-8679-test hours. Therefore, bearing lives of 2072 hours will be demonstrated by 1036 hours of normal rated power testing. The 30 x 10^6 cycles will have been demonstrated on all components with the exception of the rotor shaft with the accumulation of 1036 hours. The rotor shaft will have demonstrated 7.7 x 10^6 cycles with the accumulation of 1036 hours. A separate rotor shaft bench test at increased speed with simulated hub moment, torque, and thrust will be conducted to prove infinite life (30 x 10^6 cycles). The minimum duration of the regenerative test rig program is estimated to be nine months.

The factor of 2.0 determined by the schedule in MIL-T-8679 is considered to be conservative when compared with the missions defined in the contract specification. One test hour is equivalent to the calculated mission hours as follows:

Ferry mission - 1500 nautical miles - 9.53 hours = 1 test hr
20-ton payload- 20 nautical miles - 4.88 hours = 1 test hr
12-ton payload- 100 nautical miles - 8.56 hours = 1 test hr

(b) The Power Absorption Rig - The power absorption rig will accomplish production acceptance testing at full-speed, full-torque power. Three 5000-horsepower electric motors, mounted in parallel and equally spaced around the input shaft, will power the rig at the input power turbine connection. The torque will be absorbed at the rotor hub by three groups of two eddy current brakes connected in

series. Gearboxes will be employed to change speed from rotor shaft r.p.m. to appropriate brake r.p.m. The tail rotor drive shaft will likewise be reacted by an eddy current brake.

The cost of the test equipment is as follows:

Regenerative rig	-	$988,750.00
Power absorption rig	-	$781,620.00
Power provisions for both programs	-	$190,000.00

The costs shown above are for completely installed proven test rigs, and do not include the transmission to be tested.

It is recommended that both the regenerative and the absorption systems be employed for the following reasons:

(1) The regenerative system is for testing to MIL-T-8679 which will bring the transmissions through preliminary flight rating test, military qualification test, and endurance test.

(2) The power absorption rig will be used for production acceptance testing concurrently as the requirements of MIL-T-8679 are met.

(3) The regenerative system permits comparative testing of materials or design improvements under identical operating conditions without interruption of other related programs.

(4) The power absorption rig has within it the possibility of being connected to a rotor whirl rig with minimum additional cost.

(c) Compatibility Test Program - The compatibility testing of the major parts of the TIGR power system requires combined consideration of engine, transmission and rotor. Engine research and development time will be the pacing element of the program because of the long lead time involved (22 months through PFRT, calendar year 1967). Therefore, considerations are discussed below which recognize this situation. A logical systematic procedure is described for obtaining the TIGR system for HLH in the most economical and expeditious manner, and which can be initiated at this time.

Conceptually, the TIGR turbine would be driven by a single gas generator ducted to one quadrant. The turbine would drive an existing UH-2 helicopter gearbox through attachment at the appropriate place in the gearbox to match turbine power/speed with UH-2 gearbox power/speed at that point. An existing UH-2 helicopter rotor system would absorb the power. The following objectives would be achieved:

(1) The turbine would be developed.

(2) Quadrant end effects would be established.

(3) Clearances required by thermal growth and by asymmetric turbine loading would be established.

(4) The pivoted-vane freewheeling system would be established and evaluated.

(5) Turbine mounting stiffness would be investigated.

(6) The test rig tooth impact vibratories will be more severe than the planned HLH transmission. The test rig power input location has single-mesh tooth contact, while the planned HLH transmission has a hunting-mesh 8-planet input. The absence of any related problems in test rig operation will indicate that tooth impact vibratories will not be a factor.

(7) The rotor operates at approximately twice HLH rotor speed with approximately half the number of HLH rotor blades. This would provide a valid investigation of gas generator inlet effects from cyclic rotor downwash, exhaust effects from cyclic rotor downwash, exhaust impingement effects on rotor behavior, and exhaust/rotor downwash effects on ground clearance requirements.

(8) The engine would be developed prior to transmission development, and the test rig would be available for engine/transmission testing, with power to be absorbed by club blades.

(9) The engine and transmission would be developed prior to rotor development and the test rig would be available for rotor development as well as for engine/transmission/rotor testing.

C. COMPARATIVE EVALUATION OF POWER TRANSMISSION SYSTEMS

1. Parametric Analysis To Define Total Aircraft

Aircraft gross weight, power required, and aircraft weight empty are found by iterative process in conjunction with aircraft performance calculations. Engine weight is not varied, as engine weight in reality is a step function of the number of gas generators, requiring a power level to be reached at which three or five gas generators would be used instead of four. Parametry is used to define the weight of the components in the other categories. The specific weights of components determined by this study are incorporated in the parametric analysis. The parametric analysis predicts a design gross weight of 82,200 pounds for the lighter system. Included in this weight is a projected growth allowance of 1350 pounds. The analysis of reference 6, modified with respect to rotor, hub, body, engine, and main transmission, predicts a design gross weight of 81,800 pounds. The modified method of another manufacturer, however, predicts a design gross weight of 84,500 pounds. Although these predictions are in reasonable total agreement, details vary widely, indicating that trending data cannot be considered reliable in view of the long extrapolation from present experience. Body weight, in particular, should be investigated further. Operational considerations (cargo handling methods, armor protection, special maintainability provisions) will also have significant effect. Better definition through additional design study is required prior to final determination of HLH system effectiveness.

2. Performance

The performance is calculated for two configurations: the segregated rotor hub system and the integrated rotor hub system. This difference manifests itself in an increase in the take-off gross weight as can be seen in the weight summary of Table XI. The integrated rotor hub system weight is approximately 4400 pounds heavier than the segregated rotor hub system.

The horsepowers required for the various missions are calculated from the parametric curve supplied by USAAML and reproduced as Figure 2. This curve is cross-plotted in Figure 3 for the various mission velocities and flat plate drags for easy interpolation. These curves are the basis for determining the shaft horsepower required which, in

conjunction with the fuel flow rates, is used to calculate the fuel load needed for each mission. These fuel flows are found in Figure 4 for 4-engine operation and cross-plotted in Figure 15. This figure presents the fuel flow for 4-engine operation, 3-engine operation and 2-engine operation. The horsepowers for the engine-out operation are calculated using the method employed in determining the engine-out performance of Table II. The results of these calculations are presented in the performance summary of Table XII and Table XIII.

The ferry mission is to be calculated at the best speed for maximum range. In order to determine this, the specific air range is calculated for various gross weights as a function of velocity and then the maximum specific air range is plotted versus gross weight as are the velocities at which these maximum specific air ranges occur. These results are plotted in Figure 16 and are instrumental in calculating the fuel load for the ferry mission.

The autorotative rate of descent is calculated for the transport mission. In the heavy lift mission, the helicopter will carry its cargo externally, and the cargo must be jettisoned during autorotation. The ferry mission is STOL and with a heavy fuel load. Although specified at sea level, cruising will more realistically be at substantial altitude. During autorotation, this fuel load shall be jettisoned.

The fuel loads for the various missions are presented in Table XII and Table XIII, along with all the pertinent data for each mission.

TABLE XI

HEAVY LIFT HELICOPTER WEIGHT SUMMARY

Group Weight Breakdown	Segregated Rotor Hub Power System	Integrated Rotor Hub Power System
Rotor - Main	9278	9624
Rotor - Tail	1184	1184
Tail Surfaces	1108	1123
Body	6991	7316
Alighting Gear	2571	2755
Flight Controls	1161	1177
Engine Section	447	570
Engines	3463	3463
Engine Lubrication System	127	127
Fuel System	510	545
Engine Controls	112	112
Starting System	141	141
Main Transmission	3380	6456
Accessory Transmission, Rotor Brake, Drive Shaft	1364	1364
Lubrication System	235	235
Tail Rotor Shaft	260	260
Intermediate Gearbox (Twin Tail Rotors)	369	369
Upper Gearboxes (194 lbs each)	388	388
Lube System - Upper Gearboxes and Intermediate Gearboxes	33	33
Auxiliary Power Unit and Installation	173	173
Instrumentation and Navigational Equipment	220	220
Hydraulics	466	495
Electrical	623	625
Electronics	262	262
Furnishings	200	200
Auxiliary Gear	927	935
Projected Growth Allowance	1351	1379
WEIGHT EMPTY	37344	41536
Crew (3)	600	600
Payload	40000	40000
Fuel	3620	3900
Oil and Trapped Liquids	505	505
Miscellaneous Equipment	60	60
GROSS WEIGHT	82129	86601

TABLE XII

COMPARISON OF PERFORMANCE OF IDENTICAL MISSIONS BY TOTAL AIRCRAFT - HLH WITH SEGREGATED HUB POWER SYSTEM

MISSION		TRANSPORT*	HEAVY LIFT*	FERRY*
Design Gross Weight	(lb)	82200	82200	82200
Basic Weight	(lb)	37909	37909	37909
Crew (3 at 200 lbs ea)	(lb)	600	600	600
Operating Weight	(lb)	38509	38509	38509
Auxiliary Tank	(lb)	-	-	4570
Fuel Load	(lb)	7900	3620	59200
Payload	(lb)	24000	40000	-
Takeoff Gross Weight	(lb)	70409**	82129	102279
Load Factor		2.92	2.50	2.05
Hover, OGE, USAAML Power Required	(hp)	***	12350	-
Radius Mission:				
Average Velocity (out/in)	(kt)	110/130	95/130	-
Cruising Altitude	(ft)	SL	SL	-
Radius	(naut mi)	100	20	-
Range Mission:				
Average Velocity	(kt)	-	-	115
Cruising Altitude	(ft)	-	-	SL
Range	(naut mi)	-	-	1500
Autorotational Speed	(fpm)	2390	-	-

* Mission description is given in the section titled Program Discussion of this report. The missions are listed here to show the number of gas generators operating during the mission profile

** Maximum hover, OGE, gross weight at 6000 ft., 95°F day - 72300 lbs

*** Horsepower available, 6000 ft., 95°F - 11350 lbs

Transport:

1. Warmup and takeoff, 2 minutes at sea level, normal rated power.

2. Hover, OGE, for 3 minutes with 4 engines operating.

3. Fly out 100 nautical miles at sea-level altitude with 12-ton payload at true airspeed (TAS) = 110 knots and with 3 engines operating.

4. At midpoint hover, OGE, for 2 minutes with 3 engines operating.

5. At sea-level altitude return to base at true airspeed (TAS) = 130 knots with 2 engines operating.

6. Land with reserve of 10 percent of initial fuel.

Heavy Lift:

1. Warmup and takeoff, 2 minutes at sea level, normal rated power.

2. Hover, OGE, for 5 minutes with 4 engines operating.

3. Fly out 20 nautical miles at sea-level altitude at true airspeed (TAS) = 95 knots and with 3 engines operating.

4. At midpoint hover, OGE, for 10 minutes with 3 engines operating while unloading payload.

5. At sea-level altitude return to base at true airspeed (TAS) = 130 knots with 2 engines operating.

6. Land with reserve of 10 percent of initial fuel.

Ferry:

1. Warmup and takeoff, 2 minutes at sea level, normal rated power.

2. Fly out for a range of 1500 nautical miles at sea level in the following manner:

(a) For 332 nautical miles, fly on 4 engines
(b) For 1168 nautical miles, fly on 3 engines

3. Land at base with 10-percent reserve of initial fuel.

TABLE XIII

COMPARISON OF PERFORMANCE OF IDENTICAL MISSIONS BY TOTAL AIRCRAFT - HLH WITH INTEGRATED HUB POWER SYSTEM

MISSION		TRANSPORT*	HEAVY LIFT*	FERRY*
Design Gross Weight	(lb)	86600	86600	86600
Basic Weight	(lb)	42102	42102	42102
Crew (3 at 200 lbs ea)	(lb)	600	600	600
Operating Weight	(lb)	42702	42702	42702
Auxiliary Tank	(lb)	-	-	4800
Fuel Load	(lb)	8140	3900	61823
Payload	(lb)	24000	40000	-
Takeoff Gross Weight	(lb)	74842**	86602	109325
Load Factor		2.90	2.50	2.00
Hover, OGE, USAAML Power Required	(hp)	***	13000	-
Radius Mission:				
Average Velocity (out/in)	(kt)	110/130	95/130	-
Cruising Altitude	(ft)	SL	SL	-
Radius	(naut mi)	100	20	-
Range Mission:				
Average Velocity	(kt)	-	-	115
Cruising Altitude	(ft)	-	-	SL
Range	(naut mi)	-	-	1500
Autorotational Speed	(fpm)	2420	-	-

* Mission description is given in the section titled Program Discussion of this report. The missions are listed here to show the number of gas generators operating during the mission profile.

** Maximum hover, OGE, gross weight at 6000 ft., 95°F day - 73400 lbs

*** Horsepower available, 6000 ft., 95°F - 11350 lbs

Transport:

1. Warmup and takeoff, 2 minutes at normal rated power.

2. Hover, OGE, for 3 minutes with 4 engines operating.

3. Fly out 100 nautical miles at sea-level altitude with 12-ton payload at true airspeed (TAS) = 110 knots and with 3 engines operating.

4. At midpoint hover, OGE, for 2 minutes with 3 engines operating.

5. At sea-level altitude return to base at true airspeed (TAS) = 130 knots with 3 engines operating.

6. Land with reserve of 10 percent of initial fuel.

Heavy Lift:

1. Warmup and takeoff, 2 minutes at sea level, normal rated power.

2. Hover, OGE, for 5 minutes with 4 engines operating.

3. Fly out 20 nautical miles at sea-level altitude at true airspeed = 95 knots with 20-ton payload and with 4 engines operating.

4. At midpoint hover, OGE, for 10 minutes with 4 engines operating while unloading payload.

5. At sea-level altitude return to base at true airspeed = 130 knots with 3 engines operating.

6. Land with reserve of 10 percent of initial fuel.

Ferry:

1. Warmup and takeoff, 2 minutes at sea level, normal rated power.

2. Fly out for a range of 1500 nautical miles at sea level in the following manner:

(a) For 480 nautical miles, fly on 4 engines
(b) For 1020 nautical miles, fly on 3 engines

3. Land at base with 10-percent reserve of initial fuel.

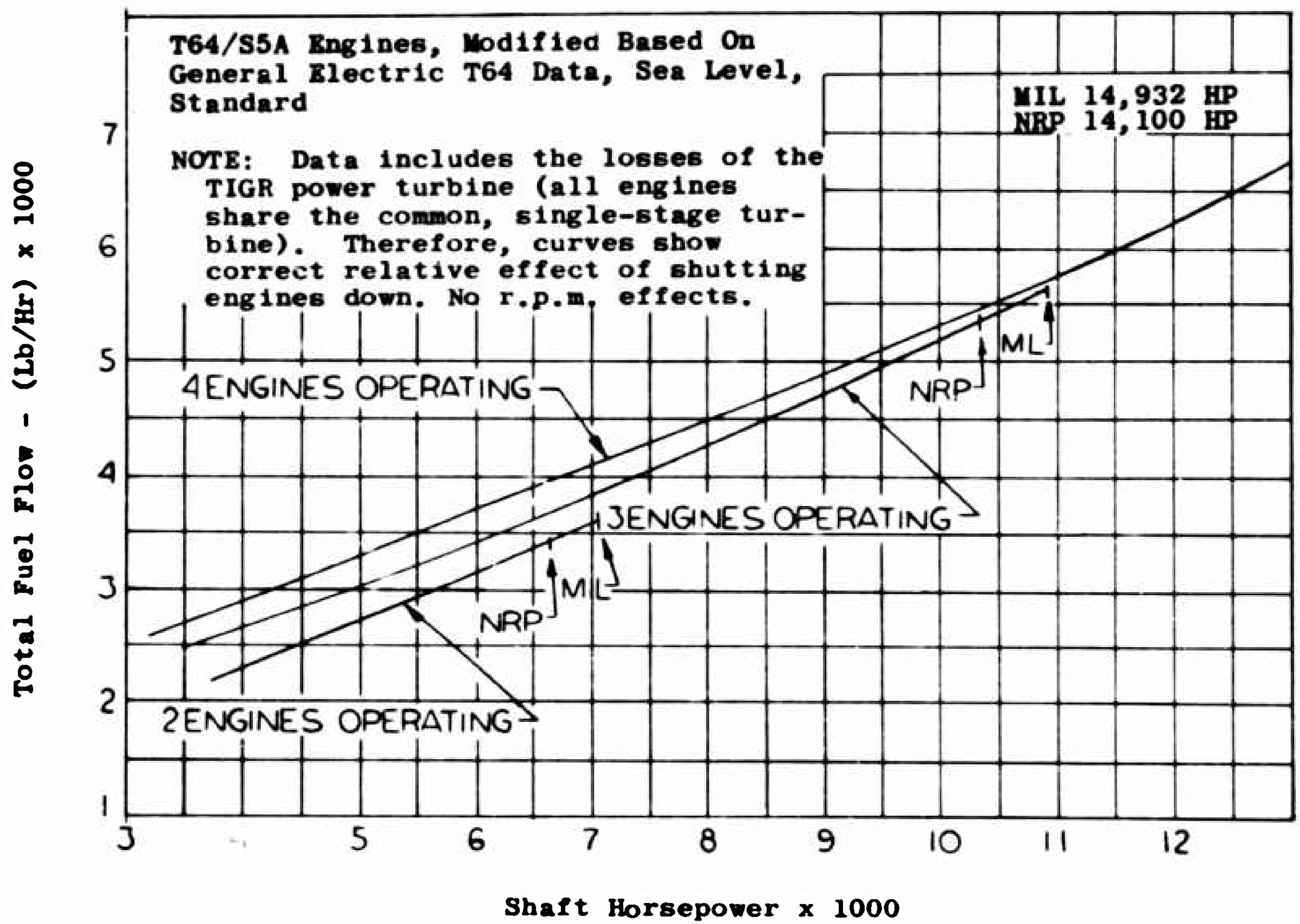


Figure 15. Fuel Flow Versus Power

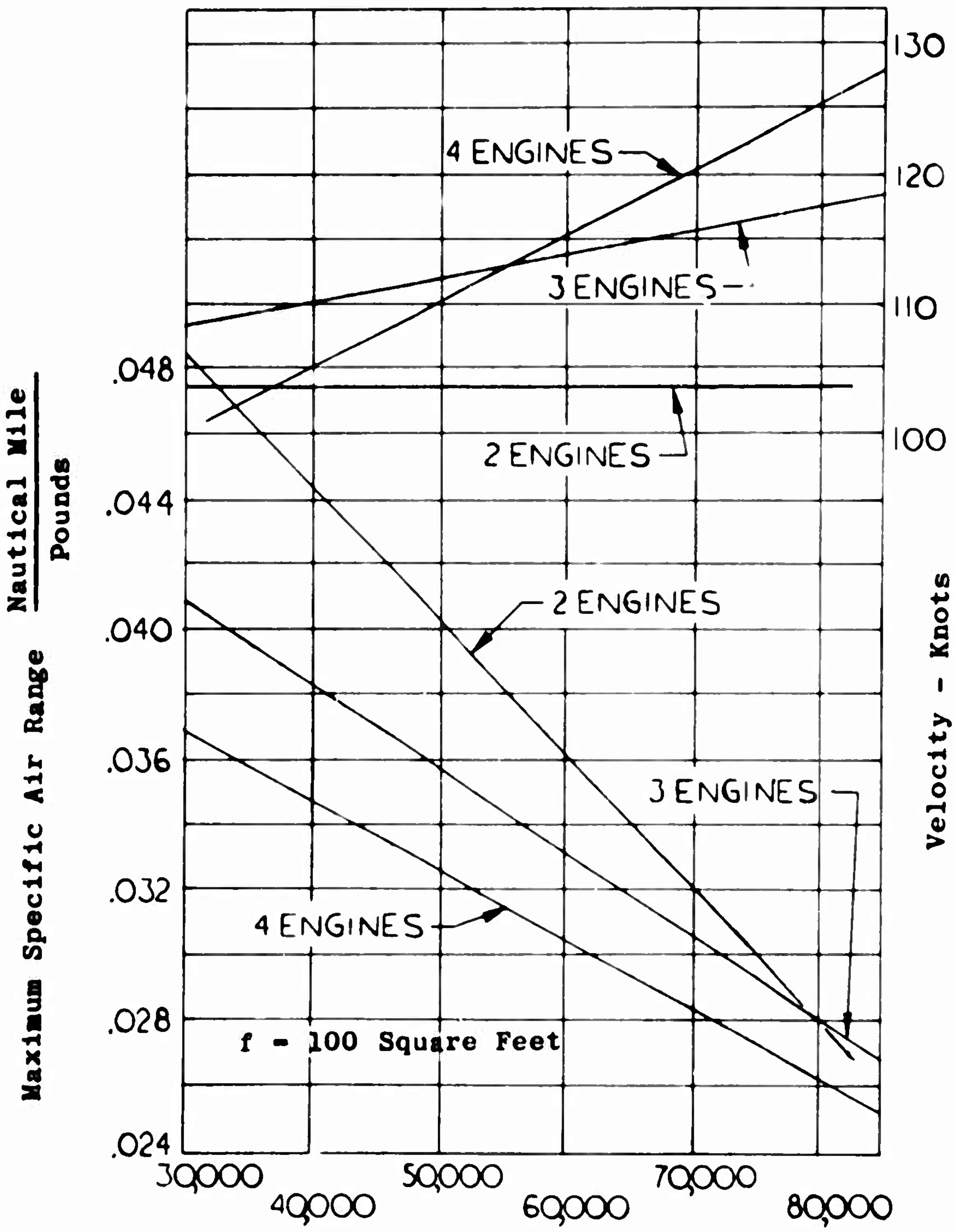


Figure 16. Specific Air Range Versus Gross Weight - Sea Level, Standard

3. Comparison of Total Aircraft

Both HLH aircraft are based on power system design to the same criteria. This has determined operating weight empty. Design gross weight was fixed by the heavy lift mission for both systems. The comparison is shown in Table XIV.

Both HLH aircraft perform identical missions. The difference in performance shows up as difference in fuel consumption. Fuel consumption weights for the three missions are shown in Table XIV below.

TABLE XIV

COMPARISON OF TOTAL AIRCRAFT - WEIGHT AND PERFORMANCE

	Operating Weight	Gross Weight	Fuel Consumption By Mission: Transport	Heavy Lift	Ferry
Integrated Rotor Hub HLH	42,702	86,600	8,140	3,900	61,823
Segregated Rotor Hub HLH	38,509	82,200	7,900	3,620	59,200
Weight Difference	4,193	4,400	240	280	2,623

For cost/effectiveness comparison of the effect of power systems on HLH, it is necessary to identify the direction of trends and evaluate opposing trends. In this case, all trends are congruous and no quantitative evaluation is required. The HLH with the segregated rotor hub system is superior. With respect to cost, the segregated rotor hub HLH is lighter and less difficult to design, develop and manufacture; therefore it is cheaper. With respect to effectiveness, three elements must be considered: performance, maintainability and reliability. The HLH with the segregated rotor hub has better performance (less fuel is used), is more easily maintained (smaller components), and is more

reliable (fewer unfamiliar components, leading to fewer design and maintenance errors).

For the development-time comparison of the effect of power systems on HLH, it is necessary to evaluate lead time required for various components. Engine development time would be the same for both systems, and has a longer lead time than transmission design and manufacture. Transmission differences are therefore not contributory, as either transmission could be made available for development testing before engine development was completed. Development time for either transmission to establish 1200-hours TBO and 3600-hours service life is estimated to be of 9-months minimum duration.

CONCLUSIONS

As a result of design, analysis, and comparative evaluation of two complete power transmission systems, including adjacent aircraft systems and resulting HLH aircraft, it is concluded that:

1. Power transmission design using the multiple gas generator/remote turbine/gearbox concept shows a 280-percent improvement in mean time between failure, where failure is defined as mission abort, over conventional multiple engine/transmission design practice for engine combination, change of direction, speed reduction, and freewheeling provision for engine-out operation.

2. An epicyclic transmission with the rotor hub segregated from the gearbox is lighter and cheaper than the alternate configuration using a rotating final stage ring gear integral with the rotor hub.

3. Locating the rotor controls within the rotor mast was found to be desirable and feasible.

4. HLH aircraft weights determined tentatively by the study for the superior system are: transport mission - 70,409 pounds, including 7,900 pounds of fuel; heavy lift mission - 82,179 pounds, including 3,620 pounds of fuel; and ferry mission - 102,279 pounds, including 59,200 pounds of fuel.

5. Insufficient data on structural weight are available to assure confident estimation of the absolute cost and performance of the complete helicopter.

6. The tentative rotor selection and the speed/drag assumptions used in the HLH performance analysis for this transmission study indicate that HLH weight and cost predictions may be reduced by additional study of HLH subsystems, because of the conservative assumptions required at this time. Also, weight and cost reduction may be anticipated to result from technological advances during the interval prior to the 1967-1972 development period.

7. No maximum-effort development appears to be required for any of the technical areas that could be investigated in this design and analytical study. Testing of a dynamic power system appears to be required to identify any interaction problems.

RECOMMENDATIONS

Recommendations are made as follows:

1. Conduct dynamic testing of a TIGR power system to investigate rotor/gearing/turbine interactions and to identify any required development areas not obvious from paper studies.

2. Conduct additional studies for better HLH system definition. Need is noted in Conclusions 5 and 6. Desirably, the studies would include rotor design and analysis, structural design and analysis, cost effectiveness analysis and configuration design and analysis. These could result in a reconsideration of mission definition, which, therefore, should also be included.

BIBLIOGRAPHY

1. American Standards Association, "Load Ratings for Ball and Roller Bearings", American Standards Association, 6 January 1959, pp. 6-12.

2. Botstiber, D. W., and Kingston, Leo, "High-Capacity Gearing", Machine Design, December 1952, pp. 130-160.

3. Cornell Aeronautical Laboratory, Inc., "Mobile Army Air Transport Study, Phase II", DDC - AD No. 234262, July 1959, pp. 36-39, 42-48.

4. Dudley, D. W., Gear Handbook, McGraw-Hill Book Company, New York, New York, 1962, Section 14, pp. 17-19, 24-26.

5. Dudley, D. W., Practical Gear Design, McGraw-Hill Book Company, New York, 1954, pp. 141-142.

6. Hiller Helicopters, "Transport Helicopter Design Analysis Methods", Report Number 473.6, 30 November 1955, pp. 59-93.

7. Kleinlogel, A., Rigid Frame Formulas, Frederick Ungar Publishing Company, New York, New York, 1952, pp. 151-153.

8. MIL-F-18372(AER), "Flight Control Systems: Design, Installation and Test of Aircraft (General Specification For)", 31 March 1955, paragraphs 1.2.1, 3.1.2.3.2, 3.1.2.3.2.1, and 3.1.2.6.

9. MIL-E-8593(ASG), "Engines, Aircraft, Turboprop, General Specification For", 3 September 1954, pg. 3.14, Figure 3 - Flight Maneuver Load Diagram.

10. MIL-T-8679, "Test Requirements, Ground, Helicopter", 5 March 1954, paragraph 3.6.3.3.

11. Schauble, J. J. and Maloney, P. F., "An Approach to Helicopter Structural Reliability and Fatigue Life", Journal of the American Helicopter Society, Volume 9, Number 4, October 1964, pp. 13-19.

12. Weidhuner, D. D., "An Army Aircraft Propulsion Forecast", Institute of the Aerospace Sciences, IAS Paper Number 63-68, 21-23 January 1963, pp. 7.

13. Jones, A. B., Analysis of Stresses and Deflections, Volumes 1 and 2, New Departure Division, General Motors Corporation, Bristol, Connecticut, 1946.

14. Jones, A. B., "A General Theory For Elastically-Constrained Ball and Radial Roller Bearings Under Arbitrary Load and Speed Conditions", Journal of Basic Engineering, ASME Transactions, June, 1960.

15. Lundberg, G., "Elastische Berührung Zweier Halbräume", VDI Forschung, Volume 10, No. 5, September/October, 1939.

16. Lundberg, G. and Palmgren, A., "Dynamic Capacity of Rolling Bearings", Acta Polytechnica, Mechanical Engineering Series, Volume 1, No. 3, Stockholm, Sweden, 1947.

17. Lundberg, G. and Palmgren, A., "Dynamic Capacity of Roller Bearings", Acta Polytechnica, Mechanical Engineering Series, Volume 2, No. 4, Stockholm, Sweden, 1952.

APPENDIX

THE ANALYSIS OF A SYSTEM OF BALL AND ROLLER BEARINGS IN WHICH THE RACEWAYS DEVIATE FROM CIRCULARITY

FOREWORD

An analytical means is developed whereby the internal rolling-element load distribution is obtained for each bearing of a system of ball and/or roller bearings. Once the rolling-element load distribution is known, fatigue life and other performance data are readily obtained.

Hand computation means cannot provide the internal load distributions in rolling-element bearings except for the simplest cases of loading and are inadequate for high-duty bearings operating under combined load.

In this analysis the entire system of bearings and shaft are viewed as a single nonlinear elastic system and the internal rolling-element loads are brought into equilibrium with the external loads acting on the shaft.

Provision is made for initial conditions such as preload, misalignments, etcetra, of any bearing.

Provision is also made for the simulation of bearing race distortions by allowing, in effect, a variable clearance at any rolling-element position.

The solution has been programmed for a high-speed digital computer.

THEORETICAL DEVELOPMENT

THE EQUILIBRIUM OF THE SYSTEM

Figure 17 illustrates a typical bearing system in which there are p rolling-element bearings on a single shaft. An angular-contact ball bearing is shown for illustrative purposes. There are t points on the shaft at which external loads act in five degrees of freedom. Forces and moments are referred to orthogonal X_t Y_t Z_t coordinate systems and are designated F_{t_1} - - - F_{t_5}. Point o is an arbitrarily chosen center of moments at which the displacements of the system δ_1 - - - δ_5 will be evaluated. Bearing reactions acting on the shaft are F_{p_1} - - - F_{p_5} and are opposed in direction to the displacements δ_{p_1} - - - δ_{p_5} of the inner ring with respect to the outer.

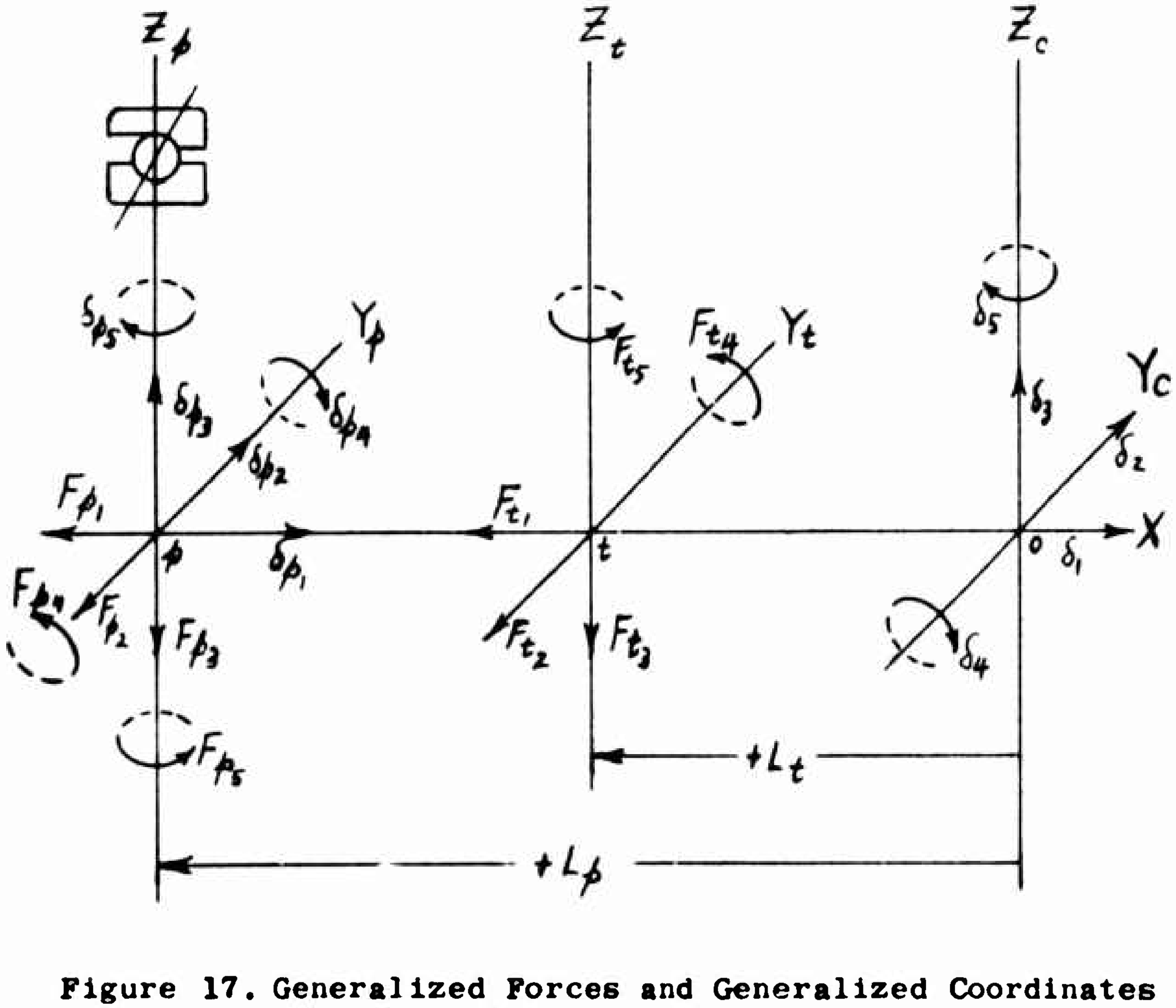


Figure 17. Generalized Forces and Generalized Coordinates of a Typical Bearing System

Equilibrium of the system requires that

$$\sum_{p} F_{p_1} + \sum F_{t_1} = 0 \tag{1}$$

$$\sum_{p} F_{p_2} + \sum_{t} F_{t_2} = 0 \tag{2}$$

$$\sum_{p} F_{p_3} + \sum_{t} F_{t_3} = 0 \tag{3}$$

$$\sum_{p} (F_{p_4} + L_p F_{p_3}) + \sum_{t} (F_{t_4} + L_t F_{t_3}) = 0 \tag{4}$$

$$\sum_{p} (F_{p_5} + L_p F_{p_2}) + \sum_{t} (F_{t_5} + L_t F_{t_2}) = 0 \tag{5}$$

These equations are nonlinear in the variables $\delta_1 - - - \delta_5$

Figure 18 defines the placement of a ball bearing inner ring and a roller bearing outer ring with respect to their coordinate systems. Figure 19 shows the positive sense of the contact angle.

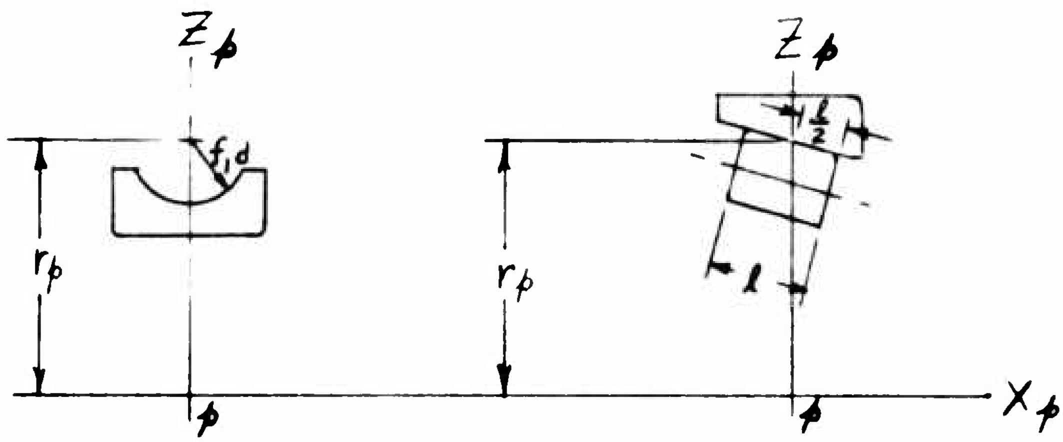


Figure 18. Bearing Ring Coordinates

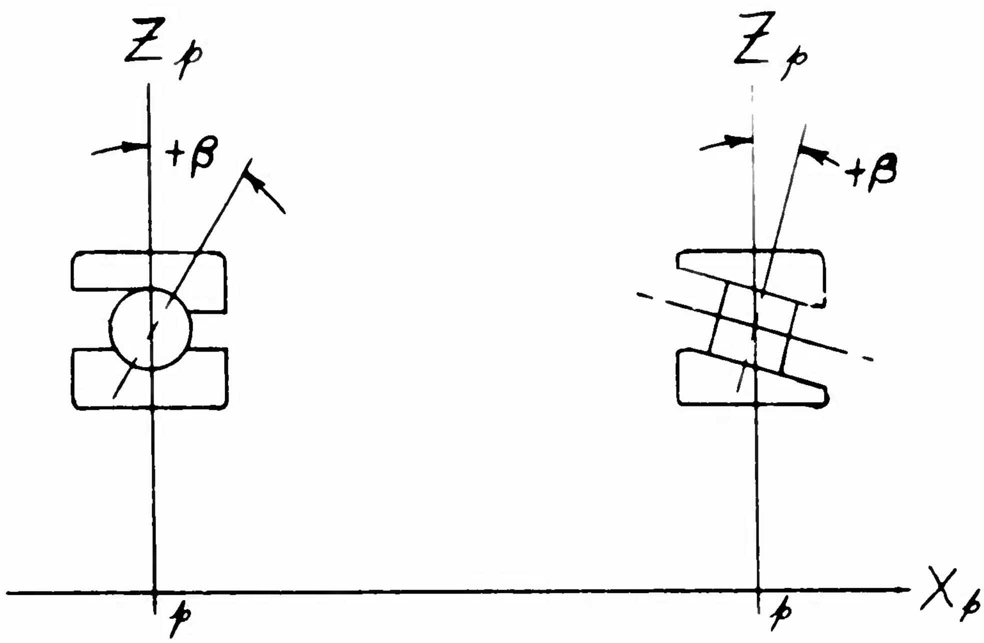


Figure 19. Rolling Element Contact Angle

The internal rolling-element load distribution depends upon the relative displacements $\delta p_1 - - - \delta p_5$ of inner and outer rings. These are directly related to the displacements $\delta_1 - - - \delta_5$ at point o.

$$\delta p_1 = \delta_1 + \delta p_1'' \tag{6}$$

$$\delta p_2 = \delta_2 + L_p \delta_5 + \delta p_2'' \tag{7}$$

$$\delta p_3 = \delta_3 + L_p \delta_4 + \delta p_3'' \tag{8}$$

$$\delta p_4 = \delta_4 + \delta p_4'' \tag{9}$$

$$\delta p_5 = \delta_5 + \delta p_5'' \tag{10}$$

$\delta p_1'' --- \delta p_5''$ are initial ring displacements.

BEARING REACTIONS

1. **Ball Bearing**

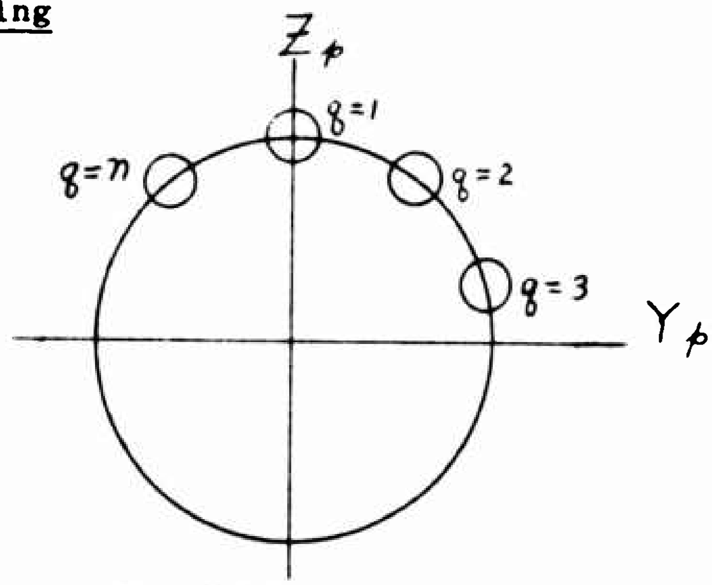


Figure 20. Rolling Element Index

If the body forces arising from the rolling elements motions are neglected the total compression of a ball positioned at g is

$$\Delta_g = B d \left[D_g - 1 \right] \tag{11}$$

B is a factor derived from the race curvatures and is

$$B = f_1 + f_2 - 1 \tag{12}$$

f_1 and f_2 are, respectively, the outer and inner curvatures as decimals

$$D_g = \left[\left[\sin\beta + \frac{1}{Bd} \left(\delta_{p_1} + r_p \left\{ \delta_{p_4} \cos\frac{2\pi(g-1)}{n} + \delta_{p_5} \sin\frac{2\pi(g-1)}{n} \right\} - \Delta_{x_g} \right) \right]^2 + \left[\cos\beta + \frac{1}{Bd} \left(\delta_{p_2} \sin\frac{2\pi(g-1)}{n} + \delta_{p_3} \cos\frac{2\pi(g-1)}{n} - \Delta_{z_g} - \frac{\Delta P_D}{2} \right) \right]^2 \right]^{1/2} \tag{13}$$

Δx_g and Δz_g are axial and radial deviations of the inner race curvature center from its proper position to permit simulation of ring distortions.

ΔP_D is any change in the bearings internal diametral clearance, as, for example, a differential expansion. A positive value of ΔP_D corresponds to an increase in clearance.

If $D_g \leq 0$ the ball is unloaded.

The operating contact angle β_q is generally different at each ball position.

$$\tan \beta_q = \frac{\sin\beta + \frac{1}{B_d}\left[\delta_{p_1} + r_p\left(\delta_{p_4}\cos\frac{2\pi(q-1)}{n} + \delta_{p_5}\sin\frac{2\pi(q-1)}{n}\right) - \Delta_{x_q}\right]}{\cos\beta + \frac{1}{B_d}\left[\delta_{p_2}\sin\frac{2\pi(q-1)}{n} + \delta_{p_3}\cos\frac{2\pi(q-1)}{n} - \Delta_{z_q} - \frac{\Delta P_d}{2}\right]} \qquad (14)$$

The contact force P_q is related to the compression of the ball by

$$P_q = K\,\Delta_q^{3/2} \qquad (15)$$

where K is an elastic constant depending on material and the geometry in the neighborhood of the contacts.

The bearing reactions F_{p_1}---F_{p_5} can now be evaluated.

$$F_{p_1} = \sum_{q=1}^{n} P_q \sin\beta_q \qquad (16)$$

$$F_{p_2} = \sum_{q=1}^{n} P_q \cos\beta_q \sin\frac{2\pi(q-1)}{n} \qquad (17)$$

$$F_{p_3} = \sum_{q=1}^{n} P_q \cos\beta_q \cos\frac{2\pi(q-1)}{n} \qquad (18)$$

$$F_{p_4} = r_p \sum_{q=1}^{n} P_q \sin\beta_q \cos\frac{2\pi(q-1)}{n} \qquad (19)$$

$$F_{\phi_5} = r_\phi \sum_{q=1}^{n} P_q \sin\beta_q \sin\frac{2\pi(q-1)}{n} \tag{20}$$

2. <u>Roller Bearing</u>

The total compression of a roller is given by

$$\Delta_q = \left[\delta_{\phi_1} + r_\phi\left(\delta_{\phi_4}\cos\frac{2\pi(q-1)}{n} + \delta_{\phi_5}\sin\frac{2\pi(q-1)}{n}\right) - \Delta_{x_q}\right]\sin\beta + \left[\delta_{\phi_2}\sin\frac{2\pi(q-1)}{n} + \delta_{\phi_3}\cos\frac{2\pi(q-1)}{n} - \Delta_{z_q} - \frac{\Delta P_0}{2}\right]\cos\beta \tag{21}$$

The elastic approach of the roll to a race is given by Professor G. Lundberg as

$$\Delta_{i_q} = \left[\frac{1}{\theta_1} + \frac{1}{\theta_2}\right]\frac{P_q}{l_e}\cdot\frac{2}{\pi}\left[1.8864 + \ln\left(\frac{l_e}{2b_{i_q}}\right)\right] \tag{22}$$

l_e is the effective roll length, b_{i_q} the semi-width of the pressure area.

θ_1 and θ_2 are elastic constants depending on material.

Where $i = 1$ for outer race and 2 for inner.

If the roller is hollow the increase in deflection of the roll is given by Timoshenko as

$$\Delta_{H_q} = \frac{P_q R^2}{A E e}\left\{\frac{\pi}{4} - \frac{2}{\pi}\left(1 - \frac{e^2}{R^2}\right) + \frac{2e}{R}\left[\frac{2}{\pi}\left(1 - \frac{e}{R}\right) - \frac{\pi}{8}\right] + \frac{3\pi}{4}\cdot\frac{E}{G}\cdot\frac{e}{R}\right\} \tag{23}$$

Where

R = mean radius of roller

A = cross-section area

E = Young's modulus

G = modulus of elasticity in shear

and

$$e = R - \frac{(d - d_H)}{2 \ln \frac{d}{d_H}} \tag{24}$$

where d_H is the inside diameter of the roll.

Then

$$\Delta_q = \Delta_{1_q} + \Delta_{2_q} + \Delta_{H_q} > 0 \tag{25}$$

The value of P_q can be found from Equation 25.

The reactions F_{ϕ_1}---F_{ϕ_5} of the roller bearing on the shaft are

$$F_{\phi_1} = \sin \beta \sum_{q=1}^{n} P_q \tag{26}$$

$$F_{\phi_2} = \cos \beta \sum_{q=1}^{n} P_q \sin \frac{2\pi(q-1)}{n} \tag{27}$$

$$F_{\phi_3} = \cos \beta \sum_{q=1}^{n} P_q \cos \frac{2\pi(q-1)}{n} \tag{28}$$

$$F_{\phi_4} = r_\phi \sin \beta \sum_{q=1}^{n} P_q \cos \frac{2\pi(q-1)}{n} \tag{29}$$

$$F_{\phi_5} = r_\phi \sin \beta \sum_{q=1}^{n} P_q \sin \frac{2\pi(q-1)}{n} \tag{30}$$

METHOD OF SOLUTION

The basic variables are the displacements $\delta_1 --- \delta_5$ at point o, for once these are found the complete rolling-element loads for all bearings in the system are easily found. Equations 1 through 5 cannot be solved directly and iterative techniques are resorted to.

An assumption is made of the variables $\delta_1 --- \delta_5$ and Equations 1 through 5 evaluated. In general they will not be satisfied and there will exist the residuals ϵ_1 through ϵ_5 .

Improved values are found from

$$\delta_{j_{n+1}} = \delta_{j_n} - [a_{ji}]^{-1} \{\mathcal{E}_i\} \qquad \begin{matrix} i = 1,5 \\ j = 1,5 \end{matrix} \qquad (31)$$

The elements of the coefficient matrix are

$$a_{ij} = \frac{\partial \mathcal{E}_i}{\partial \delta_j} \qquad \begin{matrix} i = 1,5 \\ j = 1,5 \end{matrix} \qquad (32)$$

Iteration of Equation 31 yields the δ_j to any desired accuracy.

1. <u>The Motion of a Ball in a Ball Bearing</u>

Figure 21 shows a ball with its center constrained to the plane of the paper and rotating about its own center with the angular velocity ω_B directed at α

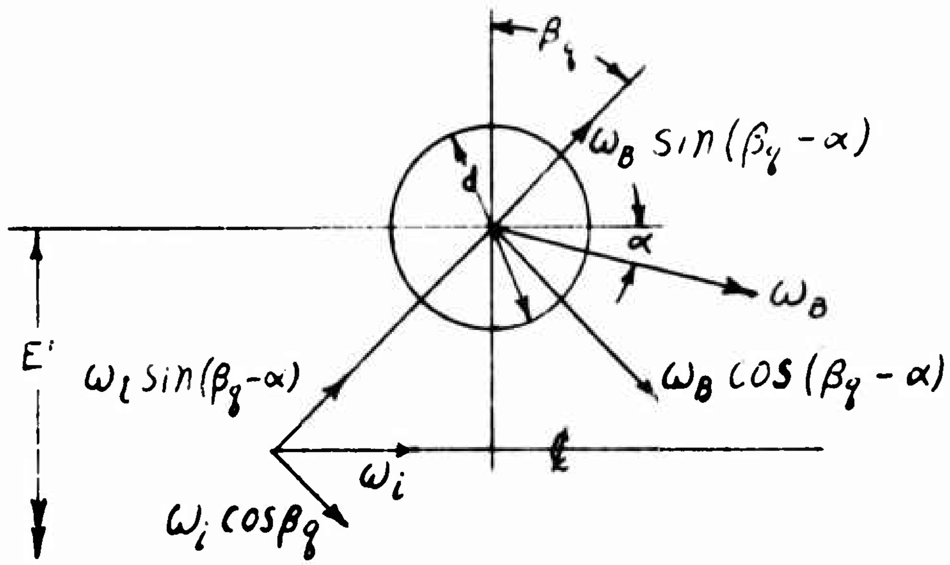


Figure 21. Kinematic Diagram of a Ball

Each race rotates with the angular velocity ω_i , where i is 1 for an outer race and 2 for an inner.

$$\omega_B = \frac{Z_i E'}{d} \cdot \frac{(1 + Z_i \gamma)\, \omega_i}{\cos(\beta_q - \alpha)} \tag{33}$$

where

$$Z_1 = 1 \tag{34}$$

$$Z_2 = -1 \tag{35}$$

$$\gamma = \frac{d \cos \beta_q}{E'} \tag{36}$$

For the outer race to be stationary, the ball must orbit with the angular velocity Ω_E such that

$$\Omega_E = -\omega_1 \tag{37}$$

Then the absolute velocity of the inner race is

$$\Omega_2 = \omega_2 - \omega_1 \tag{38}$$

$$= \omega_2 \left(1 - \frac{\omega_1}{\omega_2}\right) \tag{39}$$

From Equation 33

$$\frac{\omega_1}{\omega_2} = -\frac{(1-\gamma)}{(1+\gamma)} \tag{40}$$

and

$$\omega_1 = -\frac{\Omega_2}{2}(1-\gamma) \tag{41}$$

$$\omega_2 = \frac{\Omega_2}{2}(1+\gamma) \tag{42}$$

$$\omega_B = -\frac{E'\Omega_2(1-\gamma^2)}{2d\cos(\beta_\ell - \alpha)} \tag{43}$$

Similarly, when the outer rotates and the inner is stationary

$$\omega_1 = \frac{\Omega_1}{2}(1-\gamma) \tag{44}$$

$$\omega_2 = -\frac{\Omega_1}{2}(1+\gamma) \tag{45}$$

$$\omega_B = \frac{E'\Omega_1(1-\gamma^2)}{2d\cos(\beta_\ell - \alpha)} \tag{46}$$

For simultaneous rotation of both races

$$\omega_1 = \frac{(\Omega_1 - \Omega_2)(1-\gamma)}{2} \tag{47}$$

$$\omega_2 = -\frac{(\Omega_1 - \Omega_2)(1+\gamma)}{2} \tag{48}$$

$$\omega_B = \frac{E'(\Omega_1 - \Omega_2)(1-\gamma^2)}{2d \cos(\beta_g - \alpha)} \tag{49}$$

With α arbitrarily chosen there will be a spin of the race with respect to the ball at both contacts. The magnitudes are

$$\omega_{s_i} = \mp[-\omega_i \sin\beta_g + \omega_B \sin(\beta_g - \alpha)] \tag{50}$$

In practice the value of α is not arbitrary but depends upon the relative magnitudes of the torques required to produce spin at the two contacts.

The torque Q_{s_i} required to produce spin with an elliptical pressure area is

$$Q_{s_i} = \frac{3}{8} \mu P a_i E(k) \quad \text{pound-inches} \tag{51}$$

where

μ = coefficient of sliding friction

P = contact force - pounds

a_i = semi-major axis of pressure area

E(k) = complete elliptic integral of the second kind and is the eccentricity of the pressure ellipse.

High speed photographs of ball motion have confirmed the theory that the ball rolls on one race without spin while all spin occurs at the other. Spin occurs at that contact having the lesser eccentricity of pressure ellipse.

For spin to occur at the contact one finds

$$\tan\alpha = \frac{\sin\beta_g \cos\beta_g}{\cos^2\beta_g + Z_j \gamma} \qquad j = 1, 2 \tag{52}$$

When all spin occurs at the inner contact the ball is said to have "outer-race control" and vice versa.

The substitution of Equation 52 into Equation 50 shows that regardless of the type of control, the angular velocity of spin at the spinning contact is

$$\omega_s = (\Omega_1 - \Omega_2) \sin\beta_g \tag{53}$$

2. The Calculation of Fatigue Life

Fatigue life is calculated in accordance with the statistical theory of rolling-element bearing fatigue life of Lundberg-Palmgren (reference 17). This theory is the basis for the Anti Friction Bearing Manufacturers Association (AFBMA) methods used for hand calculation of rolling-element bearing fatigue life and the basis for most bearing manufacturer's ratings.

It can be more accurately applied here since the true internal rolling-element load distribution is known.

3. Ball Bearing

Lundberg and Palmgren (reference 17) give the capacity of a ball bearing race contact as

$$Q_{g_i} = A\left[\frac{2f_i}{2f_i - 1}\right] \frac{[1 + Z_i\gamma_g]^{1.39}}{[1 - Z_i\gamma_g]^{1/3}} \left(\frac{d}{E'}\right)^{.3} d^{1.8}\, n^{-1/3} \qquad i = 1, 2 \tag{54}$$

A is a material fatigue constant.

When $d > 1$ in. the ball diameter exponent is reduced to 1.4 in accordance with AFBMA practice.

Since, with all except pure thrust load, each ball bears a different load, it is necessary to find an equivalent rolling-element load.

The equivalent rolling-element load is that ball load, which, if applied equally at each stressing, would produce the same life as the actual, varying loads.

For a raceway which rotates with respect to load the equivalent rolling-element load is

$$P_E = \left[\frac{1}{n}\sum_{q=1}^{n} P_q^{3}\right]^{1/3} \tag{55}$$

For a raceway which is stationary with respect to load the equivalent rolling-element load is

$$P_E = \left[\frac{1}{n}\sum_{q=1}^{n} P_q^{10/3}\right]^{3/10} \tag{56}$$

The life of a raceway in hours for 90-percent probability of survival is

$$L_i = \frac{16667}{|\Omega_1 - \Omega_2|}\left(\frac{Q_{ci}}{P_{Ei}}\right)^3 \tag{57}$$

where P_{Ei} is appropriately chosen from equation (55) or (56).

The life of the complete bearing is

$$L = \left[\frac{1}{L_1^{10/9}} + \frac{1}{L_2^{10/9}}\right]^{-9/10} \tag{58}$$

4. <u>Roller Bearing</u>

The capacity of a race contact of a roller bearing is

$$Q_i = A' \frac{[1+Z_i\gamma]^{29/27}}{[1-Z_i\gamma]^{1/4}} \left(\frac{d}{E'}\right)^{2/9} d^{29/27}\, l_e^{7/9}\, n^{-1/4} \tag{59}$$

where A' is a fatigue constant depending on material, type of crowning and method of guidance of the roller.

For a raceway which rotates with respect to load, the equivalent rolling-element load is

$$P_E = \left[\frac{1}{n}\sum_{q=1}^{n} P_q^{4}\right]^{1/4} \tag{60}$$

For a raceway which is stationary with respect to load, the equivalent rolling-element load is

$$P_E = \left[\frac{1}{n}\sum_{q=1}^{n} P_q^{9/2}\right]^{2/9} \tag{61}$$

The life of a raceway is

$$L_i = \frac{16667}{|\Omega_1 - \Omega_2|}\left(\frac{Q_i}{P_E}\right)^4 \tag{62}$$

and the life of the complete bearing is

$$L = \left[\frac{1}{L_1^{9/8}} + \frac{1}{L_2^{9/8}}\right]^{-8/9} \tag{63}$$

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Multiple gas generators, installed in a conventional horizontal position, are all gas-coupled to the same peripherally-driven remote turbine of the lift and cruise fan type. The remote turbine is mounted co-axially to a speed-reducing gearbox which is also co-axial with the rotor. The concept is known as the Turbine Integrated Geared Rotor "TIGR". The TIGR arrangement eliminates from transmission design the functions of engine combining, change of direction, part of the required speed reduction from conventional engine speed to rotor speed, misalignment couplings and all of the multiple individual-engine overrunning-clutch provisions required for engine-out operation and for autorotation. A three-phase program of design, analysis, and comparative evaluation of TIGR, including the effect on adjacent systems and the resulting Heavy Lift Helicopter (HLH) aircraft, is presented. The results show TIGR has a 280 percent improvement in mean time between mission-abort failure over conventional multiple engine/transmission practice for the entire HLH power train from the engine inlets to the rotor hub. Cost per HLH flight hour is reduced, and the high fuel efficiency of mechanically-driven rotors is retained. TIGR is believed to be eminently practical and is ready for dynamic testing.

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